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SHOCK, VIBRATION AND ASSOCIATED ENVIRONMENTS

PART II

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OFFICE OF
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THE SECRETARY OF DEFENSE
Research and Engineering

The 31st Symposium on Shock, Vibration and Associated Environments was held at the Hotel Westward Ho, Phoenix, Arizona on October 1-4, 1962. The U.S. Air Force was host.

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FOREWORD

This section of the Bulletin contains the keynote addresses of the 31st Symposium. Papers dealing with the development of specification requirements and with test techniques are included, some of which were not presented at the Symposium. Panel discussions on topics most closely related to the subject matter in this volume are also included.

Suggestions to improve the Symposia and the Bulletin are always welcome. They should be addressed to Code 4021, U.S. Naval Research Laboratory, Washington 25, D. C.

W.W. Mutch.

March 15, 1963

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ADDRESS

Major General R. G. Ruegg
Commander, Aeronautical Systems Division

When Dr. Mutch invited me to talk to you this afternoon, he left the subject to my discretion.

The selection was not an easy one to make. After a review of the advance program, I felt that I should discuss an extremely light subject at this point in the Symposium, one that would be a change of pace for you in the midst of four days of technical sessions.

(Besides, I never thought it a wise policy to talk about shock and vibration to people who have just had a full meal.)

Yet, the environmental problems confronting us in the Air Force are so critical that I welcome this opportunity to discuss them. I know that nowhere could I find a more knowledgeable audience, or one which could contribute more to the solution of our problems.

Our job in the Air Force is to develop advanced aerospace systems. They include aircraft, aerodynamic and ballistic missiles launched from aircraft, ground-launched aerodynamic and ballistic missiles, and aerospacecraft.

Obviously, this range of systems encounters virtually every conceivable environmental condition and combination of such conditions.

Of paramount consideration in the development of all our systems are the factors of flight safety and reliability. And therein lies our especially deep concern about the nature of environmental phenomena encountered. Nor are these environmental problems posed solely by the flight requirements of future operations. We have them on hand today, both in quantity and in complexity.

In the case of aircraft, for example, we succeeded in increasing the fatigue life of critical aluminum forgings, which tied wing spars to fuselages. Now we are plagued by the problem of stress-corrosion, a condition caused by high stresses and accentuated by exposure to salt air or other corrosive elements.

In our constant effort to accurately predict or take into account all potentially critical environmental problems, we find that we must don bifocal glasses. We need one lens for the little picture, and the other for the big picture. What complicates our problem is that we must look through both lenses simultaneously.

Let me cite the modern manned aerospace system as a case in point.

We consider the vehicle system as an assemblage of man, machine, and structure which operates as a unit in performing its essential mission. Yet, from an environmental standpoint, we also must consider the system as an assemblage of individual parts. We must do so because the individual parts may be subjected to environments significantly different, in intensity and in form, from the vehicle itself. For example, the noise level near jet engine exhausts may be 165 decibels, while in interior locations, man and equipment may be subjected to levels of only 105 decibels. This represents a significant difference since it involves a factor of 1000 in terms of sound pressure.

Considering this and many more examples like it, it becomes obvious that we also must think of the environmental factor in terms of each specific part of the vehicle. Thus, we cannot form neatly-packaged environments for the whole vehicle or place our confidence in oversimple or quickie criteria. Such oversimplification, we feel, would compel us to pay the price in reduced reliability.

Yet, we cannot be excessively conservative in our precautions to insure the very highest levels of reliability in all directions. If we do so, we pay an equally high price in the overdesign of parts, with resulting decreases in performance or payload.

To achieve high reliability without overdesign, then, we must make every effort to

predict with greater accuracy the environmental conditions to be encountered, and the tolerance of the vehicle, its structure, and its occupants to those environments.

At the present time, the attempt to do so entails a rather extensive cycle of investigation during the vehicle's design, development, and testing phases.

First, the environmental characteristics (including time variations) are estimated for the entire anticipated life of the vehicle. Then, on the basis of this estimate, the vehicle's subsystems, components, and parts are individually designed.

Under such conditions, designing frequently becomes an inexact process. And we must look to development testing to prove the tolerance of the test articles, as well as to increase the accuracy of our original environmental estimates. In many cases, the test findings at this stage point up the need for redesign.

Even then, we must go on to test the entire first article vehicle, not only to insure that all of its parts can tolerate the combination of environments involved during its mission, but also to again check the accuracy of our definition of these environments. The latter must be done, to give us the data needed for the next redesign of any parts which failed or malfunctioned.

It sounds like a complicated, technical version of "The House That Jack Built," but it was this very same type of cycle, for example, that we were compelled to follow in order to combat acoustical fatigue in our development of the B-52 and B-58 weapon systems.

Nor in this cycle do we subject the vehicle or its parts to all environmental aspects simultaneously. One reason for not doing so is the great expense involved in building facilities capable of simultaneously duplicating combined environments throughout the entire range of their severity.

Even without these facilities, the present costs, in time and funds, of going through the cycle of development and tests for but a single environment are extremely high. In order to reduce these costs, without paying the price of reduced reliability or over-design, we must gain the capability of predicting environmental characteristics and tolerances to a much higher degree of accuracy. And to that end, we solicit your assistance.

Now let me briefly review two environmental problem areas of particular importance to the development of Air Force weapon systems—structural fatigue induced by acoustical and flight environments, and impact and shock.

First, the acoustical environment as it relates to structural fatigue: During the early years, as you know, we were concerned with aircraft noise primarily because of its effect on the hearing of flight personnel. Its cumulative effect was partial deafness. Its most immediate and obvious effect, however, was impaired communication, whether speakers or earphones were used. The quality of earphones was relatively low during that period, especially in a noise environment. And a major effort was required to hold the interior noise in aircraft to levels permitting reasonable intercommunication.

Noise levels have greatly increased over the years. The acoustical power generated by the B-52 engines, for example, exceeds that of the earlier C-54 engines by a multiple of 10. On the basis of present trends, we might well expect that the acoustical power generated by future power plants will increase by a factor of 10 every few years. This rapid increase presents the extremely challenging problem of protecting our personnel, as well as vehicle equipment and structure.

The problem of noise became especially severe with the use of high-performance engines, such as the J57 and J79. After long exposure to a sufficiently high noise intensity, for example, we discovered that delicate electronic and mechanical equipments malfunctioned or permanently failed. More unexpectedly, certain structural portions of the vehicles failed as a result of sonic fatigue. And this problem had not been adequately appreciated by aircraft designers who previously had directed their efforts towards reducing structural weight.

Considering the level of noise we expect in our future operation, the problem of acoustically induced fatigue is one which merits our concentrated attention. Indicative of the importance we attach to this problem is the new acoustical facility now being constructed at Wright Field. It will provide a test environment of 1-million watts of acoustical power, or more than 100 times the acoustical power produced by one of the earlier sonic fatigue facilities completed in the middle 1950's.

The cost of this facility will exceed $8 million, and it becomes obvious that the financial

burden involved in attempting to keep pace with the requirements of our future aerospace vehicles will be an almost prohibitive one. All of us, therefore, must redouble our efforts to find other methods to get this job done, whether by use of reduced scale models or by other means.

In this same structures area, the repeated loading of aerospace structures, that is, low-frequency fatigue, still remains a major obstacle to flight safety.

The progress we have made has been encouraging, but much more remains to be done. For example, if we are to have reliable, lightweight, flight vehicle structures with an adequate service life, we must know the magnitude and occurrence frequency of flight loads and environments, the distribution of the loads throughout the structure, the strength and life characteristics of construction materials, and, most important, we must be able to accurately predict structural life as a function of operating hours.

We must develop the analytical tools and procedures necessary to process mass-data so that meaningful interpretations in terms of structural life can be made. We must know, for example, just how old our flight vehicles are, how many "fatigue birthdays" they have had, and how many more they can go on to celebrate. Gaining this kind of knowledge is the principal objective of the Air Force's structural integrity program.

To that end, we presently are working on the development of a compact, lightweight instrument which will be able to continuously record flight loads over a period of time covering an entire aerospace mission. By no means have we solved all the development problems involved. But, someday, we hope to install this instrument in a percentage of all first-line aerospace vehicles. The data will be processed and analyzed in automatic data processing facilities. And it will be used, not only to strengthen structural design criteria, but also to determine loading histories. By this means, we also may be able to make rational estimates of residual structural life.

Now let's turn briefly to the problem of shock and impact, more specifically to the area of dynamic loads, the equivalent of the shock environment in flight vehicle operation:

During World War II, we required maximum performance from our aircraft, and then some. As a result, serious failures occurred in the horizontal stabilizers and landing gear. Studies and tests revealed that dynamic landing loads were responsible, and a new facet of structural dynamics was emphasized. Environmental data and statistical information were needed to characterize landing speeds and attitudes. And also required were analysis methods and test data to provide design criteria and to validate approaches.

As a test pilot toward the end of World War II, I recall taking part in a program to obtain data on aircraft landing loads. After I had made an especially hard landing. I asked the project engineer: "Was that a hard landing?" His reply then and many times later was: "Yes, but make the next one harder!"

Perhaps this indicated the zeal of our environmental data-gatherers, or the eternal contest between theory and practice.

In any event, the investment in these tests and studies paid off in methods for rectifying the problems on the flexible landing gear and aircraft components and for effecting reasonable corrective actions.

As it was then, however, our real need now is to be able to anticipate and prevent problems, to minimize retrofit actions, and to assure successful performance of the complete system. To that end, we have sharpened design analysis and prediction procedures through research and weapon system information. And we have been able to define environments, such as vertical landing or sink speeds, the roughness of runways, and so on, by means of significantly improved statistical information obtained from newly developed devices.

We have supported the development of such instruments as an ultrasonic, automatic, all-weather, rate-of-descent measuring device and an automatic-surveying cart which rapidly measures the varying heights of runways and taxiways.

Together with the National Aeronautics and Space Agency and the Federal Aviation Agency, we are investigating the use of flight simulators to define sink speeds and their statistical nature for more advanced aircraft, such as the supersonic transport,

Thus, we have a continuous program underway to define environments, design criteria, specifications, and methodologies for dynamic load problems of advanced vehicles. This program will extend into the aerospace era where new dynamic load problems will occur due to docking, rendezvous, interconnected bodies, and landings on new and poorly-defined planetary environments.

In yet another direction, shock and vibration aspects also come into play when nuclear weapons effects are involved. Here, the objective might be to define the behavior and vulnerability of aerospace vehicles and to develop hardening effects. Nuclear explosions provide short-duration and high-impulsive loads, such as pressures or blasts, heat and radiation inducing spallation, fracture, and dynamic buckling in structures. These structural effects, which are relatively new in the engineering sense, but much older in the research field, were recently the subject of a special Aeronautical Systems Division Symposium—"Structural Dynamics Under High Impulse Loading." The Symposium contributed greatly to the interchange of knowledge in an area of special significance to our defense program.

Finally, let us consider environmental problems as they directly relate to two of the advanced aerospace systems we are managing at the Aeronautical Systems Division—first, the X-20 Dyna-Soar manned space glider, and second, the GAM-87 Skybolt air-launched ballistic missile.

To be boosted into orbital flight by a Titan III booster, the winged X-20 Dyna-Soar glider will be brought back to earth in a gliding maneuver under the pilot's control. One of the initial objectives of this program will be to determine a pilot's ability to re-enter the earth's atmosphere from space in a winged, maneuverable craft, and land it completely under his control.

The name Dyna-Soar is derived from Dynamic and Soaring. It means that the vehicle will use both centrifugal force and aerodynamic lift.

Centrifugal force will sustain the glider when it attains orbital speed, about 18,000 miles per hour. At this speed, it will be flying just fast enough to offset the pull of earth's gravity.

The glider will remain in orbit like a satellite until the pilot decides to return. Then, after he fires retro-rockets to decrease its orbital speed, the glider will enter the earth's atmosphere in a long glide.

The craft's wings will give it aerodynamic lift and maneuverability as it descends through the atmosphere. And the combination of high speed, extreme altitude, and maneuverability will permit the pilot to shorten or lengthen his glide by thousands of miles, and to maneuver far to the right or left of his flight path to reach his landing site. Landing the Dyna-Soar should be no more complicated than landing a modern jet fighter or the X-15 research airplane.

Parts of the surface of the Dyna-Soar will be heated in varying degrees from 2000 to 4000 degrees Fahrenheit. The pilot will remain comfortable in a cockpit kept at room temperature. The air in front of the glider—the stagnation area—will heat up to 20,000 degrees or more. This super-hot air, or plasma, will behave differently from air as we know it. Flowing back over the craft as it re-enters the atmosphere, it will appear luminous, much like a shooting star blazing across the sky.

The Dyna-Soar will be constructed of high-nickel-alloy steel, molybdenum or columbium, and ceramic materials highly resistant to heat. Unlike nose cones coated with an ablative material which can boil off, the Dyna-Soar glider will radiate heat from its surfaces back to the atmosphere.

The nose cone of an intercontinental ballistic missile plunges back into the atmosphere in a matter of seconds and must endure even higher temperatures, although for a relatively short time. In contrast, Dyna-Soar will return in a more leisurely manner and will take a longer time, over 30 minutes, to dissipate its heat.

Early X-20 Dyna-Soar flights will be made at more than 20 times the speed of sound and will last for more than 1 hour. They will provide us with a means of conducting research and development in a true flight environment.

Most important from our standpoint, Dyna-Soar will point up the dominant role that heating can play as a complicating environmental factor. And it will emphasize the need to obtain system reliability of the highest degree without the penalties of over-design.

Our GAM-87 Skybolt ballistic missile also presented us with unusual problems related to the environmental factor. This missile was designed to be launched from bombers of the B-52 type. Following a ballistic trajectory above the atmosphere, it was planned to travel at hypersonic speeds to a pre-determined target some 1000 miles away.

Unusually severe environments of noise, vibration, and shock required our special attention to assure equipment and accessory reliability. And since the missile was to be carried on a mother aircraft, it required a longer service life and significantly increased fatigue-resistant characteristics. As a result, specifications were revised and qualification tests were made to update procedures and to develop criteria which took into consideration both the vehicle and its environment.

Noise and vibration, resulting from repeated engine run-up and take-off operations of the carrier aircraft, provided the most severe environment for the entire missile, as we initially expected and later verified by measurements. The re-entry portion of the missile was expected to encounter its most severe environment due to aerodynamic turbulence and buffeting. Our efforts to reduce size and weight focused on fatigue of skin panels and light structure, and protection for the small equipments against the shock and vibration environments.

Insofar as the future is concerned, we feel that shock, vibration, and associated environments will play an ever greater role in the design of our aerospace vehicles.

We will have to expand our test facilities. We will have to obtain environmental data on a more accurate and timely basis. And we will have to firm up our design rules for obtaining a given tolerance in vehicle parts.

Above all, we will have to insure that we get the greatest possible returns from the resources given us to get the job done.

In the final analysis, our success will depend, as it always has, on how well we apply our combined talents to the task at hand.

* * *

Section 1

ENVIRONMENTAL PROBLEMS OF THE NEXT TEN YEARS

ADDRESS

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Despite progress over the past 2 decades toward solving natural and man-made environmental problems affecting Army materiel, equipment, personnel, and operations, the effort must be increased and continued during the next 10 years. Attrition, deterioration, dissipation of effort, as well as loss of time and money, continue to place the role of environment on a par with control of potential political enemies. Effective control of, or adaptation to, natural and induced environment may well be considered as the measure of achieving imbalance of power between two opposed military forces. The Army considers its major R&D problems to be firepower, mobility, communications, and survival, although upon examining these areas closely, one immediately sees that each of these functional areas resolves down to materiel and design to cope with limiting effects of some form of natural or induced environment. Therefore, one might be permitted to paraphrase the Army R&D problem as: the design of equipment and techniques by means of which our military personnel can control large or small scale combat operations in any natural or induced environment.

The variability of environment in character, intensity, and duration is every bit as complex as the innumerable choices of secret plans by an enemy to control his defenses and offenses. It is logical to assume, however, that through scientific understanding the probabilities of predicting and outwitting environmental opposition is potentially greater because neither secrecy nor volition are inherent in the physical principles by which the inanimate environmental enemy behaves. Why then has our task to impose our control over the opposing forces of nature been so difficult? Have our designs of equipment and techniques been lacking adequate environmental criteria? Has our effort to understand the scientific nature of environment been too primitive? Have we left as a calculated risk too much to chance? Have our designs been increasing in complexity and vulnerability faster than we've learned to control the newly induced environments they create? Or has communication between the students of environment and the designers of military materiel and techniques been imperfect? Perhaps it is some of each.

Let us briefly examine the fundamental nature of shock, vibration, and related environments to see where our future research effort can best be applied to the problems of environmental control.

If we reduce the problem of coping with random impulses to its least common denominator, we must assume that the responses are in the molecular or atomic structure of matter. These fundamental particles have apparently a limited set of mechanical, chemical, thermal, and electromagnetic responses manifested in change of size, motion, and reaction time. Scientific laws of adequate practical application make our understanding of these reactions for any induced impulse fairly predictable. Reaction time between an induced impulse and a molecule or atomic particle seems to be the critical stage involved in responses leading to problems of practical control over the random effects of environment. We may assume that

there are three levels of reaction time of fundamental particles which frame the problem area:

1. Reaction times sufficiently fast to absorb the energy of induced impulses without damage to materials.

2. Reaction times so slow that their inertia to impulses produce no apparent measurable response or progressive deterioration of materials.

3. Reaction times generally falling in a range between 10^{-3} and 10^{+3} seconds where responses (within the greater range of 10^{-10} and 10^{+10}) produce accumulative deterioration of materials.

Reaction time here includes response, relaxation, or apparent resistance to any impulse whether it be physical motion, thermal, electromagnetic, or chemical in nature.

Our greatest success in dealing with problems of shock, vibration, and other environmental impulses seems to have been where we have successfully introduced fast reaction time absorbers of energy on the one hand, such as pneumatic, hydraulic, and elastic systems (i.e., as in tires, transmissions, and breaking and cushioning processes). On the other hand, we have made advances through strengths of material, design, miniaturization, and immobilization of materials to increase the advantages of slow reaction time. Translation of shearing strains along lubricated planes to energy absorbing points has also been notably successful, although inventive genius in fast reaction time junctures at the strain joints between rigid materials has been less apparent. As a layman in this audience of design specialists, I may be simply exhibiting my naiveness when I predict that in the 10 years ahead the greatest advances in the field of shock and vibration may be expected in the designing of fast reaction time molecular junctions connecting rigid frames subjected to high "g's." Our progress also must be directed to the elimination of intermediate reaction-time phenomena which weaken molecular bonds of materials whether it be by introducing faster or slower reaction time fundamental particles selectively within heterogeneous systems subjected to stress and strains.

To make progress in scientific control over environmental impulses, deteriorating military devices and techniques, it is essential that the designers know the character, frequency, and duration of the impulses emanating from the environment. The students of natural environment have tried to improve their descriptions of environment along these lines; however, they have had insufficient guidance from designers as to the kinds of impulses that matter most or the sorts of environmental criteria that are most essential. The designer may feel equally lost by the physical geographer's terminology assuming that it is wholly inadequate and inapplicable to his engineering formulae.

Apropos of this situation and my earlier remarks it may be well to recall the words and guidance expressed by Lieutenant General Arthur G. Trudeau, the recent Army Chief of Research and Development, when he addressed the 30th Symposium on Shock and Vibration. He stressed two questions which I find myself here today attempting to discuss. He asked you then: "What is the true nature of natural and induced environment, and how can we more closely simulate them to remove postulation and guess?

The second question was: "How do we make the geographer, who is our expert in natural environment, and the engineer-scientist, who studies our induced environment, talk a common language?" He also stated that if you do both, we will have gone a long way to a real control of our designs in the ultimate area of use.

A year later I find myself here today as a geographer trying to breech this language barrier with you engineer-scientists. Perhaps some advantage may be gained if I attempt to analyze how we students of the natural environment have been attempting to solve our part of this mutual problem. We physical geographers and related geophysical scientists inherited systems of observational techniques which have, as in the case of meteorology, provided a climatic network over much of the globe; from the geologists and physiographers we obtain land-form descriptions; from the biologists we derive ecological patterns; and from other geophysical science we assemble existing data from glaciology, geomagnetism, oceanography, and so on. In short, the geographic environmental specialist deals with a large number of scientific disciplines; and, through his principle tool — maps — he attempts to depict the character, intensity, distribution, and temporal factors of the earth sciences. We who have fostered Army environmental research have assumed that if the R&D designers and testers of materials and equipment could establish the performance rating of line items, factor by factor, or field prove them at well-instrumented test sites, then we could, through terrain evaluation, analogues, and environmental analysis, tell the military operations personnel where over the world the items in combat will function, or otherwise, have

known limitations. Our test sites over the past decade include the Yuma, Arizona site for desert testing; Greeley, Alaska, Churchill Manitoba and Greenland sites for cold weather and polar testing; Panama site for tropical testing as well as the other proving grounds and test board sites variously located around the United States to round out the temperate zone means of evaluation.

Although notable progress has been made in this system, designed to link test performance of military equipment with predicted performance elsewhere in the world, it alone appears inadequate to solve our requirements for the next decade. Our Army R&D goal is to provide the combat units with end items that will function as intended no matter where or in what natural environment they are placed. The inability to give this assurance stems from factors which lie in the roots of our deliberations here this week. Let us examine some of the problems further, and see if we can hope to do anything about them.

The military requirements for new development items provide a limited and usually inadequate set of environmental design criteria. Loopholes are provided and generally used like a sieve; i.e., in the course of development a waiver of requirements for meeting environmental criteria may be granted if development costs are seriously increased, development time is lengthened, or the item becomes too heavy, bulky, or interferes with other design criteria. The approvers of environmental waivers generally reason that the bulk of items will not be subjected to extreme environments and are, therefore, not essential. If the designers incorporate new features and capabilities into their new end items, the users tend to develop new techniques which promptly tend to subject the item to new environmental stresses not contemplated by the designers. Environmental extremes are conservative and statistically valid, but to many they seem unreasonably severe. Actually they have to be severe because items which undergo successful tests by trained engineers will not perform equally as well in the hands of less capable users and under the inadequate maintenance resulting from combat conditions. Production models are sometimes not as well tooled as preproduction test items, therefore "environmental-slippage" occurs. For a long time there has been a design and testing goal of -65°F for many Army items. No one seriously expects that these items will be subjected to combat at this temperature minimum. If, however, the designers were to aim at a more practical goal of -20° to -40°F, which is not an uncommon range of winter temperature in parts of North America, Europe, and Asia, then experience leads us to believe that many items when poorly maintained in combat by average trained personnel will begin to show environmental deficiencies at about 0°F.

These are practical considerations; however, another real and more scientifically difficult aspect is also present. Even if we assume that designers and geographic environmental specialists are in good communication over the meaning of specific design temperatures, wind, humidity, surface hardness, and configuration, we find that in practice the combinations of individual factors in endless variations tend to create new and different environmental stresses. Temperature and wind in various combinations vary the rate of heat loss or gain. Wetness or ice on a slope reduces friction. Rain on clay reduces bearing hardness and introduces stickiness or slickness.

The geographic environmental specialist is aware of these variables and has been trying to deal with combinations of environmental factors; but, he soon runs into combinations and permutations so complex that either his data is inadequate or the resultant possible variations become too complex to map by his traditional methods.

The designer is also aware of these endless variations of combined environmental factors. He may also get lost in the earth scientist's definitions because they are not couched in terms applicable to his design formulae. If I understand the R&D designer's needs properly, he essentially wants to know the nature of environmental impulses, their intensity and duration in order to match them to the reaction times of molecular and atomic materials he has chosen to meet other military requirements.

It is here that I place the most critical problem facing us in the field of vibration, shock, and related environments for the next decade. There is no key presently available to translate environmental impulses, as used by designers, which can match the earth science data in a manner that permits world-wide prediction of performance of end items. It is not likely that either the designers or the environmental specialists can bridge this gap by themselves. It may be possible that we, working together, can establish a key which permits translation over this awkward communication barrier. A few specialists from both sides of the communication barrier need to work closely together for a sufficient time to develop an environmental design criteria "Rosetta stone;" the key I envision will originate from the basic nature of physical, thermal, chemical, and biological impulses

which produce molecular or atomic reactions. These impulses must then be organized into tables of significant ranges which the physical scientists and design engineers may already have reasonably in hand. Next, this table must be analyzed through research to determine how they can be related to the global environmental data; i.e., temperature impulses of finite intensities and duration must be interpreted into the climatologists' temperature and solar distribution and duration iso-lines. The physicists' measures of gravitational response to random shocks and vibration encountered over uneven surfaces must be translated through some common denominator to an equivalent of the physiographers' terrain roughness descriptions of world landscapes. What I suggest is no light or easy task, and would give rise to meaningful new research which may consume the energies of a team for much of the decade ahead. The results, however, would hopefully be that R&D designers and program management would have global evaluations of the distribution of natural environmental stress directly additive or subtractive to self-induced environments within any end item produced by R&D.

A step in the right direction might be, for example, to reproduce natural environmental data on absolute scales. Would a physical scientist more readily understand temperature limitations if he were given design criteria in degrees Kelvin? It might seem so for he could, without translation, comprehend that absolute extreme surface temperatures of black bodies in calm conditions may attain heat impulses of about 370°K for 4 hours at a time, or in the polar regions with maximal negative radiation the same black body surface could go to an absolute minimum of about 190°K. It would tell him at once that in the latter case there is a 40-percent deviation toward absolute zero from normal room temperatures, or in the former case, on the hot side, a 20-percent increase over the same standard room temperature. Therefore, if the range of operation of a device is limited to a 1-hour impulse of ±10 percent on the absolute temperature scale, to either side of room temperature, he knows very well that there are naturally occurring temperature ranges on the surface of the earth where the device may be inoperative. Unless drastically limited, the geographer could tell these designers the mean Kelvin temperature ranges for specific durations that a device should be able to withstand in any given theater of military operation.

On the other hand let us look at another means of approach by which the physicists, chemists, and engineers could contribute to the better understanding of logisticians, combat requirement developers, and geographers. The physics handbook and other sources of reference provide us by elements and compounds with the properties of materials affected by temperature, humidity, pressures, and other physical or electromagnetic impulses commonly generated by natural and induced environments. These lists are cumbersome if one wishes to find what happens to substances at or beyond any specific level of impulse. It might, therefore, prove helpful to reorganize these lists in sequential form showing how an increased impulse such as temperature causes fundamental property changes. The usefulness of such well-organized lists I believe would, at a glance, flag the phenomena changes which might contribute to malfunctions when design criteria are established. Design criteria ought to be made with full scientific understanding of effects not by resorting to a blind calculated risk. For years, the military have vacillated over selecting specific environmental design criteria based on climatic analysis. Designers have complained that they are too heavily taxed by imposed goals, but have they really placed their arguments on the table with full scientific logic? I hope that I've made my point clear for achieving mutual understanding across an apparent scientific language barrier.

At this point I have consumed most of my allotted time without saying too much directly related to my assigned topic. When I was first asked to address this audience, I accepted, thinking I'd be using the geographers connotation of the term environment. I quickly learned that your organization uses the term environment more in the induced sense than in the natural sense that I was accustomed to. As spokesman for the U.S. Army here today, I called for assistance and soon had a foot-thick stack of background material and suggested topics for inclusion. I do not hesitate to admit that some submissions were so good that I might easily have accepted one and made my preparation easy. In fact, there was a superb 10-page listing in one case of the functional impact of natural and induced environments giving the Army design problems now and for the 10 years ahead.

In review of the background I found, however, that the problems of shock, vibration, and related environments were basically a continuation of unsolved problems of the last decade. There were no startling new environmental problems envisaged as the Army proceeds to enhance its major capabilities in firepower, mobility, communications, and survival. I decided to spend my time, therefore, discussing

the research needed to unblock slow progress in applying fundamental science to the old problems.

In addition to gradual progress across the board in meeting environmental stresses, the Army has been performing and will continue to perform research in the human factors aspect of shock and vibration, as well as other environmental stresses. It is one thing to have perfection in equipment reliability under environmental stresses of every sort, and quite another to assure ourselves that the user or encapsulated operator is equally effective. Whereas most Air Force and Naval operating forces perform their combat functions from their transport vehicles, the Army with some exceptions must move to its combat site and dismount to perform its combat mission. If shock, vibration, and other stresses deteriorate the soldier's fighting ability by the time he reaches the critical moment, he may not be at optimum effectiveness. The studies by the U.S. Army and in the United Kingdom are trying to assess the quality and quantity of this deterioration. They, in the next few years, will be able to influence equipment design criteria, on the one hand, in an effort to minimize induced fatigue upon transported personnel and still, on the other hand, not overlook potentials of a biological and chemical nature that can keep or bring the combat soldier close to optimum efficiency at the critical time.

Human engineering has been making increasing impact upon equipment design and can be expected to increase its influence in the years immediately ahead. Let us hope that in this field, also composed of other disciplines and other technical languages, a translation key in terms of reaction times and responses to molecular behavior can be bridged between the inanimate and animate structures brought into combination.

In summary, may I add that in the 10 years ahead of problems concerning shock, vibration, and environmental stress, the U.S. Army expects simply an exponential rise in its current problems due to increasing complexity and sophistication of its new R&D. The present problems will alter little, but they will intensify as we seek assurance of freedom from environmental failures on the battlefield. Shifting emphasis from total to limited warfare potential will tend to emphasize the need to prepare for quick precise movements of troops and equipment from one type of natural environment to another. We must be prepared, as has long been our goal, to meet the endemic stresses peculiar to deserts, tropics, and frigid zones. We must be prepared to operate with equal effectiveness on or off roads, in planes, on mountains and in valleys, as well as on seashores and among rivers, swamps, jungles, and grasslands. We need to move faster, communicate more readily and always win our engagements with minimum losses. New weapons must function as planned, and we must survive the lethal aggression of both the enemy and the environment.

This blueprint may seem old and faded, but the gross requirements are not altered. The solutions we seek from the designers of our Army equipment is liberation through science and advanced technology from failures of our equipment and logistics in a new and unfamiliar geographic setting. If, in truth, as I have suggested, progress toward this goal is in part due to imperfect design criteria, perhaps at this very symposium we can lay the groundwork for guiding the research needed in the next decade that will match the design engineers' language with that of the military geographers and human researchers, so that those who must establish design criteria will be certain that the end product can perform effectively wherever it is destined to go.

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SPACE VEHICLE ENVIRONMENT AND SOME NASA FACILITIES FOR THEIR SIMULATION

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INTRODUCTION

Before this symposium ends, the first 5 years of the exploration and exploitation of space flight, which opened with the launching of Sputnik I on October 4, 1957, will come to a close. In that period of time manned space flight in near-earth orbits became a reality and the United States established a program of space exploitation for this decade that includes landing a man on the moon and returning him to earth. The nature of the total environment for space vehicle systems that must be provided for is becoming increasingly clear. If we are to build such systems sufficiently reliable to launch man into space, to cruise in space for extended periods of time, to land on the moon or the planets, and to return through the atmosphere, it is necessary that there be available on earth the capacity to duplicate these environments as closely as possible. Such ground facilities will provide a basic understanding of the design problems associated with these environments and lead to systems of increased efficiency and reliability.

In this brief look at some of the important problems arising from space vehicle system environments, it is convenient to regard them in three phases: The launch phase in which most of the problems are associated with the dynamic response of the vehicle to engine and aerodynamic inputs; a space phase in which the problems are associated with the radiation and meteoroid particles that the earth's atmosphere and magnetic field have protected us from; the atmospheric reentry phase in which the major problem is that of dissipating the tremendous kinetic energy of the vehicle by a means which will reduce the heat input to the surface to tolerable levels.

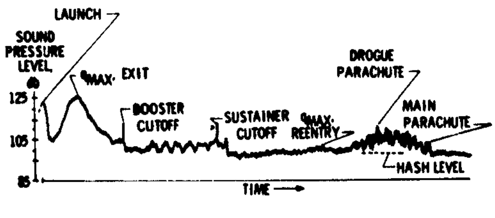

Fig. 1 - Time history of internal overall noise level for Big Joe flight

LAUNCH PHASE

Noise

During the launch phase the maxima of the noise environments around the space vehicle are generated during engine burning and during transit of the atmosphere in the region of maximum dynamic pressure. These maxima are well illustrated in Fig. 1, which displays a time history of the sound pressure levels measured inside a Mercury capsule during a suborbital check-out flight. The launching vehicle was an Atlas. Data are shown for the launch and subsequent free-flight operations of the vehicle with some of the more significant events such as launch, exit maximum dynamic pressure, booster and sustainer cut-off, maximum dynamic pressure during re-entry, and deployment of drogue and main parachutes. It should be noted that for this flight the maximum sound pressure levels occurred during maximum dynamic pressure and are of aerodynamic origin, and that the secondary maximum occurred during engine burning at low altitudes.

It is anticipated that two trends associated with engine noise will continue. These are illustrated in Fig. 2 which shows the trend of sound pressure level and the frequency at

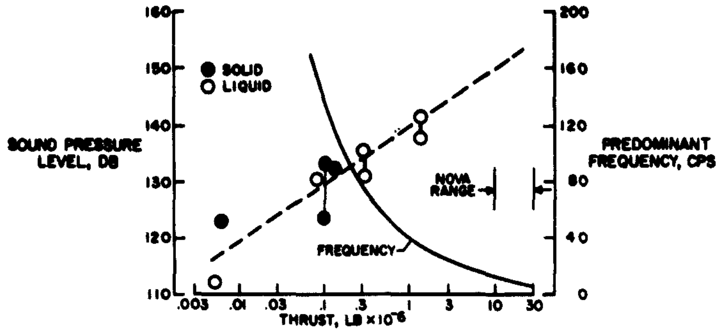

Fig. 2 - Sound pressure levels and predominant frequencies at 1000 feet from rocket engines

which most of the noise energy is concentrated, as a function of the thrust level in millions of pounds. Data are available for both solid and liquid engines and have been normalized to a distance of 1000 feet from the engine. Two aspects of the figure should be noted. The proposed Nova engine can be expected to raise the noise level at 1000 feet by 10 to 15 db over existing engines which have noise levels of about 140 db, a noise level capable of damaging ground structures not designed to resist such pressures. The second aspect is that the frequency at which the noise from Nova will peak is in the subaudible range below 20 cps. The effect of such low-frequency noise on aerospace and ground structures is for the most part unexplored.

Sound pressure levels at various distances from the engines are given in Fig. 3 for the Atlas, Saturn, and Novas of 12- and 20-million-pound thrust. It is anticipated that sound pressure levels from Nova, corresponding to those for existing engines, will be extended about 10 times their present distances. If a level above

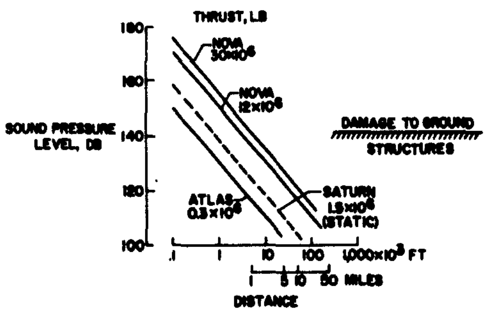

Fig. 3 - Noise levels as a function of distance for large vehicles

about 140 db is considered damaging to conventional ground structures, it can be seen that an area 1 mile in radius is included.

Current estimates of the aerodynamic noise that can be expected in flight for the Apollo command module are summarized in Fig. 4 in which sound pressure levels are plotted as a function of frequency during several extreme flight conditions. The noise levels at maximum dynamic pressure are largely aerodynamic in origin and peak at about 50 cps, and drop off at the higher frequencies as is characteristic of aerodynamic noise.

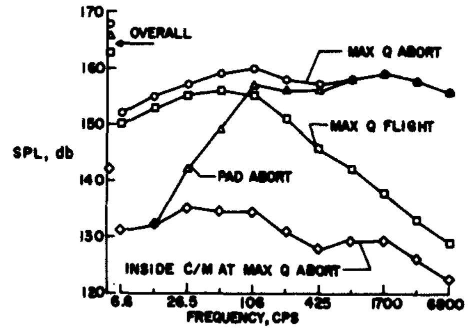

Fig. 4 - Sound pressure levels as function of frequency for Apollo command module during various flight phases

Should the rocket engines that abort the capsule be fired at maximum dynamic pressure, the highest total sound level pressures for the flight are anticipated. The most significant change in the noise level is due to the engine noise contribution at the higher frequencies.

This engine noise contribution at the higher frequencies is more clearly observed in the noise levels estimated for pad abort in which the noise is almost solely from the engines. During this operation there is little or no over-all vehicle velocity.

Noise levels inside the capsule are expected to be reduced 25 to 30 db as is indicated on the lowest curve for the maximum dynamic pressure abort conditions.

Fortunately, for the ground simulation of noise environment for space vehicles, Langley Research Center placed in operation, some years ago, the 9- by 6-foot supersonic Mach 3 blow-down tunnel shown in Fig. 5. This tunnel is operated from a large tank farm with a capacity of 1/3-million pounds of air at 600 psi. From the wind-tunnel test section the air is exhausted through a diffuser section with a 12-foot-diameter exit. The jet produces an intense noise field with over 3-million watts of acoustical power. Surveys of the noise field indicate a maximum noise level of 162 db, and within a several-hundred-foot radius of the exit, the levels are in excess of 140 db. The spectrum is relatively flat in the near field, approximating engine noise, and in the far field decay of the high-frequency components yields a spectrum more like those due to aerodynamic noise measured in flight.

The tunnel is normally used, for research on aerospace structures under both aerodynamic heating and loading, on a daily basis. Runs of the order of 10 to 40 seconds are made. The total noise exposure time and experiments made available from this tunnel operation are large compared to the launch noise and flight exposure times that are possible with high-thrust launch vehicles.

Dynamic Loads

Early next year, the Langley Research Center plans to put a new dynamics research laboratory into operation. A plan of this laboratory is given in Fig. 6. The basic feature of this

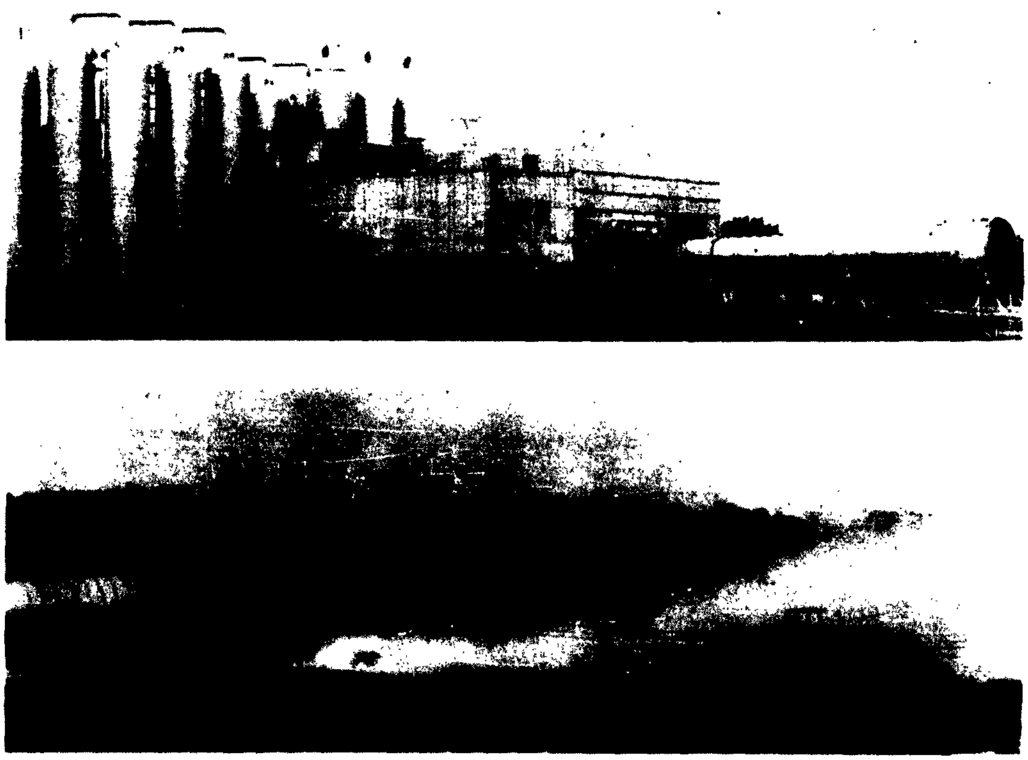

Fig. 5 - Langley 9- × 6-foot thermal structures blow-down tunnel

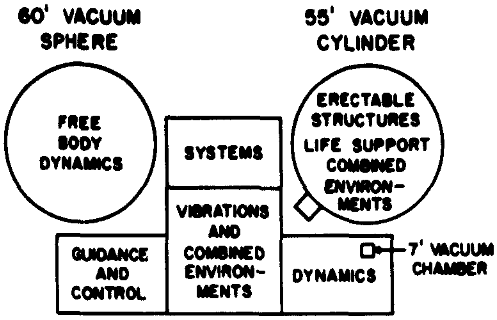

Fig. 6 - Plan of dynamic research laboratory

laboratory is that it permits a number of environments to be combined with the dynamic loading tests. Two large vacuum tanks are included, one of which is to be employed in research work on space guidance and control systems. The other is a 55-foot vacuum cylinder and is to be used for combined vacuum and dynamic loading. A smaller chamber, 7 by 7 feet, will also be available for more complex environments.

A cross section of the 55-foot cylinder is shown in Fig. 7. The centrifuge located below the floor is capable of applying a steady 100 earth "g" loading to a 500-pound test article. A 12-cubic-foot volume can be handled with only small gradients of the steady acceleration. A 3000-pound random-force shaker is mounted on the rotating arm and these forces can be superposed on the steady ones. The vacuum that can be maintained is 10^{-4} mm of Hg. The facility has been designed so that pressures corresponding to 25,000 feet altitude can be restored in 10 seconds, and sea-level conditions in 30 seconds, in order that tests in which men are involved can be run with safety.

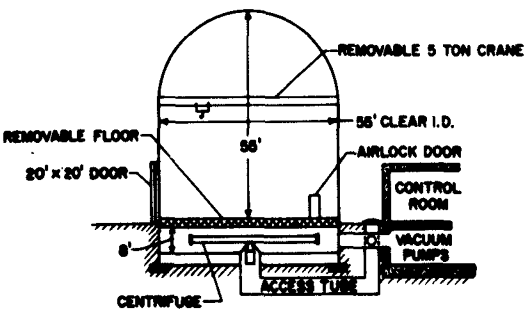

Fig. 7 - Langley 55-foot environmental research vacuum chamber

The smaller vacuum chamber is shown in some detail in Fig. 8. Test articles supported on the backstop can be loaded by a shaker of 2000-pound capacity in the range of 5 to 5000 cps. The vacuum chamber which encloses the test article is capable of being evacuated to 10^{-8} mm of Hg. In addition, quartz lamp heaters can maintain test specimen temperatures of 2000°F, and a cyropanel at liquid nitrogen temperatures can maintain a temperature as low as -320°F. A special design problem was the seal between the shaker and the chamber walls; this has been solved by using a neoprene diaphragm.

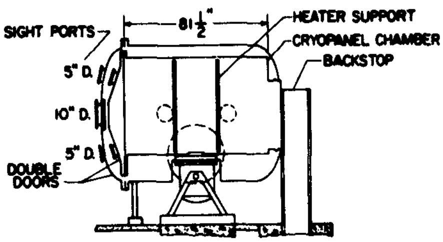

Fig. 8 - High vacuum combined environmental chamber

Also under consideration for possible use in this laboratory is a simulator which can be used to reproduce the impact and subsequent relative motion of space vehicles on lunar or planetary surfaces for which the gravitational force differs from that of earth. This simulator is illustrated in Fig. 9 and consists essentially of two masses M_1 and M_2 coupled together through a pulley and cable system. By suitably adjusting the masses the systems acceleration can be added to or subtracted from the earth's gravity. Impact between a payload and a surface can be made and the subsequent motion, relative to the impacting surface, studied.

SPACE PHASE

Flight in space presents a number of new environments with which the vehicle designer must contend. The vacuum of space, which in itself is a factor to be considered, permits the relatively free passage of charged particles, electromagnetic radiation, and the macroscopic particles which are normally observed as meteorites. The electromagnetic radiation can alter surface properties, but mechanical properties of the bulk material are easily protected by relatively thin metallic coatings. For example, the ECHO satellite, which was made of an organic polymer Mylar which is readily

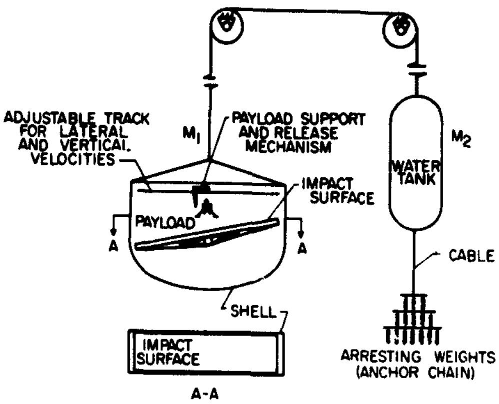

Fig. 9 - Proposed impact simulator for lunar and planetary gravitational fields

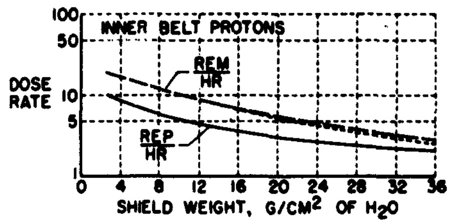

Fig. 10 - Maximum radiation dose rates in center of spherical shield for van Allen Belt protons

affected by ultraviolet, was adequately shielded to prevent material properties loss during its lifetime by a fraction of a mil of aluminum. It is, however, the elementary particle radiation and the meteoroids which constitute the more serious hazard and for which manned spacecraft must be designed.

Particle Radiation

There are three categories of particle radiation that must be considered; the relative significance of each depends on the mission of the spacecraft. They are the radiation trapped in the earth's magnetic field, the solar flare protons which are sporadically ejected from the sun and which may be encountered by a vehicle in space, and the galactic cosmic rays.

There are a number of units by which the amount of ionizing radiation and the doses absorbed are measured, but for the purposes of this discussion, the roentgen, rad and rep may be considered approximately equal. A unit, the rem, is defined; it may be regarded as an adjusted value of rep to compensate for the different biological damaging effects of the various kinds of radiation. Since this presentation is limited to protons in the range of 0.1 Bev and higher, the adjustment factor by which rep is increased to obtain rem is less than 1.5. The average lethal dose for whole body radiation of a human is 450 rem; 100 would produce sickness. Solid-state devices and the more sensitive organic materials are damaged by doses in excess of 10^6 rep.

Figure 10 summarizes the influence of various shielding thicknesses on the dose rate inside a vehicle traversing the most intense portion of the van Allen radiation belt. The ordinate is dose rate and the abscissa is shield weight in gram/cm^2 of water. If the scale of the abscissa were increased by a factor of 2, it would correspond closely to the more practical dimension—pounds per square foot. These numbers indicate that for rapid traversal of the belts, of the order of fractions of an hour, as do vehicles of the Apollo mission, little damage can be expected even to humans for small amounts of shielding. They indicate, however, that an earth orbiting vehicle, especially man-carrying, should avoid this region by remaining in close-in orbits of 100 to 200 miles. Communication satellites, which of necessity operate at altitudes of a few thousand miles must remain in these belts, and can, especially on their surfaces, accumulate in their design lifetime doses damaging to such devices as solar cells.

Space flight beyond the protection of the earth's magnetic field can expect to encounter, on a chance basis, clouds of solar protons. Figure 11 summarizes the limits of total dose behind various shield thicknesses for encounters with the three types of flares that have been observed. Data on more recent flares do not differ substantially from these. Encounters with flares of the August 22, 1958, type can occur once a month. Encounters with either high-energy or high-flux flares are less frequent, perhaps one in 4 years for the high-energy event and four in 1 year for the high-flux event. For flights of a week or more during periods of solar activity, a significant chance of an encounter exists. Protection of human occupants, that is, for doses of the order of 25 rep or less per encounter, requires total wall weights of about 12 grams/cm^2, if we consider the lower levels of estimates applicable.

In free space, whether or not there are any protons due to a solar flare, there is a steady background of galactic cosmic rays. The cosmic-ray intensity is given (Table 1) as 2.5

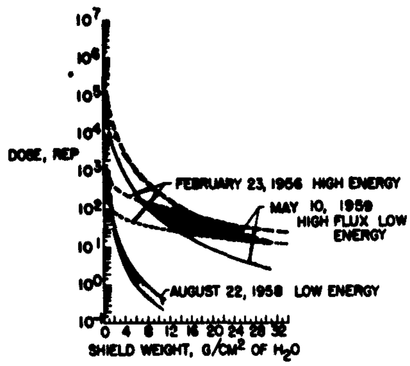

Fig. 11 - Limits of total doses in center of spherical shields for several solar flare proton events

TABLE 1
Galactic Cosmic-Ray Intensity and Dose Rate During Years of Solar Activity

Flux (only primary particles)	Overall Ionization (no outer shield except body self shielding)
$2.5 \frac{\text{Particles}}{\text{cm}^2 \text{ sec}}$	$0.45 \frac{\text{rem}}{\text{week}}$

particles/cm^2 sec and the surface dose for no external shielding has been reliably estimated at 0.45 rem per week. Such radiation is extremely energetic and not much protection is afforded without large shielding thicknesses. For long flights of the order of years limitation of the doses from these cosmic rays controls the design shield weights.

The Langley Research Center has proposed for construction in Fiscal Year 1963 a Space Radiations Effects Laboratory. This facility will permit us to produce on the ground, space radiation in the form of energetic protons up to energies of 600 Mev. A sufficient flux is available to irradiate large-scale elements of space vehicles or components in a few hours, corresponding to a year in the belts. Also in the laboratory will be a linear accelerator capable of producing electrons in the 1 to 10 Mev range.

Meteoric Particles

The meteoroid hazard is one which has been found in recent studies to be the controlling factor in the design of the details of the structural surface of a manned, earth-orbiting, space station. The meteoroid flux on which this study was based is summarized in Fig. 12. The ordinate is in impacts per foot square day, the abscissa mass in grams. The upper estimate is the most recent due to Whipple, the lower to Watson. They are based on optical and photographic observations and represent extrapolations in the lower mass range. Also shown are the latest direct measurements in space observed by microphone and break wire detectors.

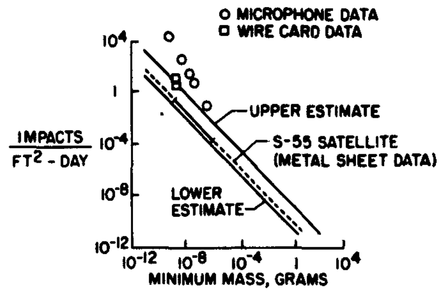

Fig. 12 - Comparison of estimated and observed meteroid fluxes

It is significant that the microphone data are trending downward and it is yet to be established how much further this trend will continue as the mass increases. The dashed curve represents a probable distribution inferred from the S-55 satellite launched from Wallops Island, Va. It was in orbit only 3 days, but its instrumentation was operable. This satellite which contained a large number of thin-walled pressure vessels sustained no puncture during the flight. The lack of punctures implies a distribution closer to the lower estimate of Watson. On the basis of a penetration estimate which is based on experimental data carried out at velocities of 23,000 fps, a velocity only one-fourth the average velocity of meteoroids, the penetration statistics for the aluminum walls of the space station were determined as function of thickness. These results are summarized in Fig. 13 and indicate that at a 50-percent probability of no penetrations of the surface of the space station wall in 1 year, thicknesses in the range of 0.040 to 0.125 inch were required.

Studies were also made of the relative weight of structure required if some punctures are allowed. These results are given in Fig. 14 which shows that significant weight saving can be made if some punctures can be tolerated. For the same probability of 50 percent, allowing five punctures a year requires only one-half the

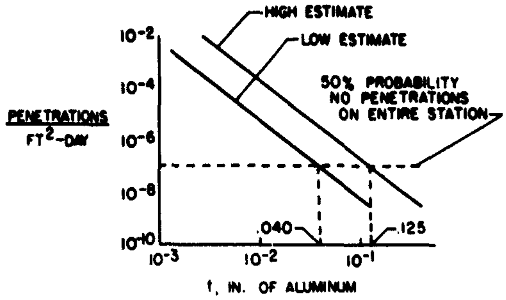

Fig. 13 - Statistics of penetration of aluminum surfaces

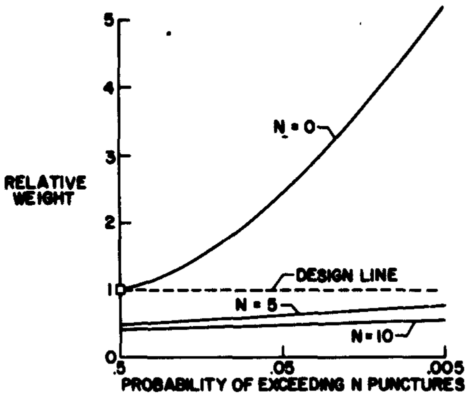

Fig. 14 - Influence of expected penetrations on relative weight of vehicle surface

wall weight. At the small probability of 0.005 of exceeding five punctures, the weight is considerably below that of the reference structure.

Simulation of meteoroid impact and penetration in ground based facilities is at present inadequate except perhaps in the range of the smallest particles. Figure 15 compares the velocity of meteoroid particles, as a function of mass, with the velocity obtained in the various present ground facilities. It is apparent that except for particle masses less than about 10^{-10} grams, attainable laboratory velocities are considerably less than meteoroid velocities. Practical interest for penetration of structure lies in the range of 10^{-2} to 10^{-4} grams and clearly in this range our present velocity is inadequate. Because of this fact penetration statistics in space are to be obtained by the NASA S-55 satellite series. However, it is still difficult to obtain information as statistically significant as is desired because of the limitation

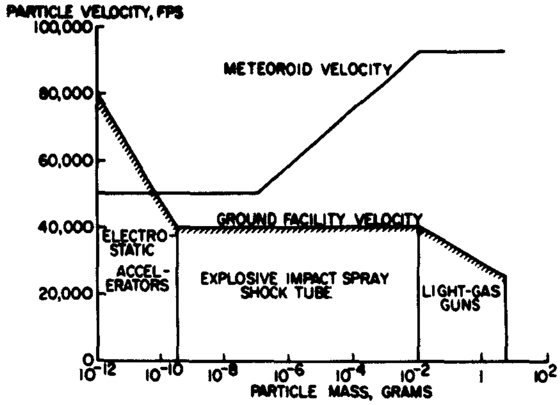

Fig. 15 - Comparison of ground facility velocities with meteroid velocities

of the area of test surfaces that can be orbited.

RE-ENTRY PHASE

Radiation Heating

The most critical aspect of design for re-entry into the earth's atmosphere is the design of the shield to protect against the heat generated by the high speed. The lowest speeds at which unpowered space vehicles can return to earth and enter the earth's atmosphere are summarized in Table 2. These speeds correspond to minimum energy deep-space missions, which require long excursion times, for example, 10 years for a trip to Saturn, and 98 years for Pluto. The latter time could be cut to about 10 years by increasing the entry velocity from 53,000 to 140,000 ft/sec. In general, therefore, we would like to consider velocities from about 40,000 ft/sec upward to, say, 100,000 ft/sec for future vehicles.

That the tremendous energy to be dissipated at such entry speeds raises large questions of survival is illustrated in Fig. 16. Consider first our current space vehicles with re-entry speeds ranging up to 36,000 ft/sec for Apollo. This speed corresponds to energy levels up to about 27,000 Btu/lb, which is higher than the heat released by a pound of gasoline, and is 5 to 10 times the heat per pound that can be absorbed by the best current heat-shield materials. Survival of our current designs thus depends on the fact that only a minor fraction of the total heat generated is actually transferred to a blunt-shaped vehicle, the rest appears as atmospheric

TABLE 2
Minimum Earth Re-Entry Velocities from Various Space Missions

Mission	Minimum Re-entry Velocity (ft/sec)
ICBM	22,000
Near-Earth Satellite	26,000
4000-Mile-Orbit Satellite	30,000
24-Hour Satellite	34,000
Lunar	36,000
Venus	38,000
Mars	39,000
Mercury	45,000
Jupiter	47,000
Uranus	52,000
Pluto	53,000
Sun's Surface	95,000

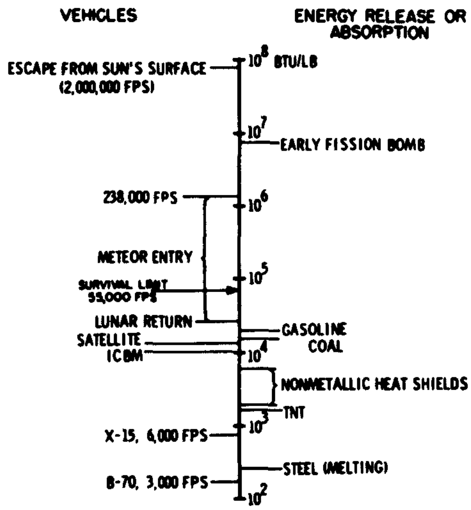

Fig. 16 - Comparison of kinetic energy per pound for various missions with energy released or absorbed in various energetic processes

heating through the action of shock waves. If we look at still higher speeds, we are impressed by the survival limit for natural meteorites, about 55,000 ft/sec. Man will have to make large improvements to exceed this limit if survival at 100,000 ft/sec is to be achieved.

The heating problem at higher-than-Apollo speeds is dominated by a complex new heat-transfer mechanism which greatly increases the heat-load fraction that must be accepted by the vehicle. If only the familiar convective heating had to be designed for, the achievement of 100,000 ft/sec would be clearly possible with current heat-shield materials. This is illustrated by the dashed line in Fig. 17; the "Heat Load" is that part of the total initial kinetic energy that must be accepted as vehicle heating for a blunt Apollo type body. In fact, however, additional heat in the form of radiation from the glowing gas cap must also be accepted. This is unimportant for Apollo, but at 100,000 ft/sec it will amount to some 10 times the convective heat load. An upper limit for the heat load of about one-fifth of the initial kinetic energy is reasonable from consideration of energy left in the wake, radiation away from the body, and other secondary effects. If we compare this heat-load limit with the heat absorption of a candidate material, graphite, we note the ability to reach a speed of about 80,000 ft/sec, for which the body would have to consist entirely of heat-shield material. This preliminary result is by no means final for two important reasons: first, the actual level of radiant heating, currently the subject of intensive research, is in large doubt, and it may be considerably less than in this upper-limit estimate; second, the effects of changes of shape and weight during re-entry, which were ignored in the above estimate, tend to reduce the relative heat load.

Thus research and more precise analysis may yet make possible practical space probes that can be designed for velocities twice as high as those of the surviving natural meteorites. Manned systems will almost certainly have to be limited to lower speeds (less than about 50,000 ft/sec) because of the intolerable decelerations which develop at the higher speeds, even when modulated lift is used to minimize this problem.

A facility on the ground to reproduce the radiation input to re-entry shapes by virtue of the exact simulation of airflow does not exist at present for the testing of materials for heat shields. Because of this limitation of the present ground facilities, thermal protection systems are being tested by using a radiant energy source in conjunction with the convective heating of an airstream heated by an electric arc. At the Ames Research Center of NASA an arc image furnace has been coupled to a heated jet by the arrangement illustrated in Fig. 18.

Because of the difficulty of making ground based experiments at extreme velocities, an advanced NASA flight re-entry research project designated Project FIRE provides for two late 1963 launchings of multiexperimental,

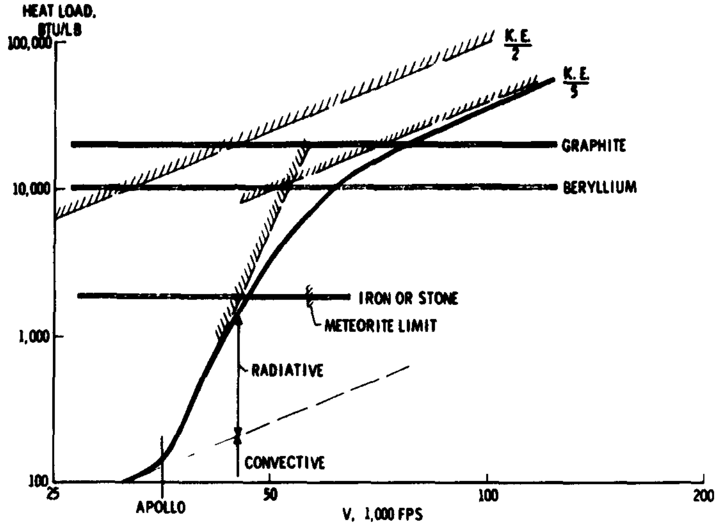

Fig. 17 - Heat load absorbed as a function of reentry velocity

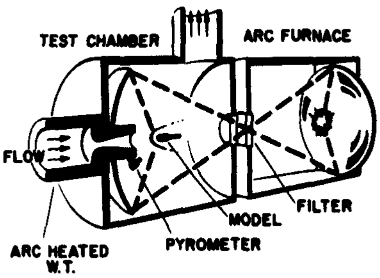

Fig. 18 - Entry heating simulator that includes radiation input

nonrecoverable spacecraft to ballistically reenter the atmosphere at an earth referenced velocity of 37,000 ft/sec. Atlas D launch vehicles will be used, with the final velocity increments provided by an Antares II stage. At the expected equilibrium stagnation temperature of 11,000°K, the blunt Apollo-shaped reentry package will be exposed to radiative heating rates of the same order as produced by convective heating. The primary purpose of the project is to obtain reliable measurements of the aerodynamic heating in the hyperbolic velocity environment, and thus provide needed guidance in the assessment of data from ground facilities, and of theoretical prediction methods. The sum of convective heating and absorbed radiation will be measured during the first half of the heating pulse, near peak heating, and on the decaying side by successively exposing beryllium calorimeter heat shields. During these periods direct measurements of the gas radiation, both spectral and total; will also be made over a sensitivity range of at least 7 decades. The spectral distribution will be measured at wavelengths from 0.2 to 0.6 micron, and the integrated radiance will be obtained from 0.2 to 4.5 microns. Afterbody temperatures and static pressures will also be measured. Over 300 data channels including temperatures, radiation intensities, body motions, and pressures will be obtained. Extensive use of ground-based radar and optical equipment is planned to provide a definition of the ground observables associated with re-entry at hyperbolic velocities.

CONCLUDING REMARKS

This brief review has presented some of the more significant environments anticipated for aerospace vehicles at present and for the future. Where possible, ground facilities have been or are being made available to explore their influence on vehicle design. There are, however, some environments not yet adequately simulated on the ground and in these instances flight programs have been started. It is anticipated that the combination of these ground and flight efforts should place the NASA research centers in the position of adequately supporting this country's present and future space program.

ADDRESS

Dr. R. O. Burns
Scientific Advisor to the
Deputy Chief of Naval Operations (Development)

The invitation to me, for this talk, specified that I should try "to focus attention on the nature of the environmental problems likely to be facing the defense services within the next 10 years."

This, of course, brings us face to face with the problem of long range planning. There are two fundamental approaches to this problem. First, from the point of view of the scientist: What things can be done to extend our knowledge of the universe, and what can be done to augment our know-how for the performance of identifiable military functions? Second, from the point of view of the military operator: What types of missions can be expected to be required, in what environment must our forces operate, what kinds of functions must be performed in accomplishing those missions, and what is the "minimum acceptable level" of performance for each class of function?

Both of these approaches must be pursued, in concert, if we are to produce a comprehensive, logical plan. I consider that the two are of equal importance, and further that they can be undertaken in isolation, that is, without direct reference to each other.

In considering the first point of view, one can develop a sense of proportion for the future by looking into the past. When the first Shock and Vibration symposium was held in January 1947, nuclear weapons, guided missiles, gas turbines, and helicopters were in their infancy. Nuclear propulsion, hydrofoil craft, ground effects machines, and systems such as POLARIS, ASROC, and SUBROC had not been born. The innovations of the past 15 years are, indeed, quite remarkable. One becomes humble when asked to suggest what innovations may occur during the next decade.

Nevertheless, I suggest that operational predictions appear to have a primary role because they establish the general vector directions along which science must proceed if we are to have the necessary know-how available before it is needed. Accordingly, I propose to explore, with you, the second point of view.

The military operate through the use of organized forces. Hence, attention must be centered on complex man-machine systems.

Parenthetically, allow me to emphasize the fact that the men are always the most important components of these systems. I exhort you never to forget them in planning your work — even though your particular assignment may seem to be oriented wholly to hardware problems.

In this context, I am defining a man-machine system to be a complex combination of men and machines, operating in concert, for the accomplishment of a specified function.

In order to analyze such systems in an orderly manner, one needs a concept for structuring the problem. A consideration of forces-in-conflict shows that there are a basic set of five functions that can be used to specify an acceptable structure. These functions can be described as follows:

1. Surveillance — The acquisition, processing, and clear-picture display of information pertinent to (a) where are my friends and what are they doing? (b) where are my enemies and what are they doing? (c) where am I? and (4) what is the environment in which we all are immersed? The output of a surveillance system is the input to a command-control system.

2. Command-Control — comprises situation assessment, decision-making, and orders pertinent to the military operation. At a level of force units and above, command-control involves decisions on tactics and strategy, nothing more.

3. Nullification — the application of force, either active or passive, for the destruction,

downgrading, or neutralization of enemy actions. Nullification systems include weapon systems, jamming and deception systems, and so on.

4. Delivery — the means for the placing of own elements or units in a desired place and orientation at a specified time. Delivery systems provide mobility; therefore, they include all vehicles and devices, as entities, that move in two or three dimensions of space and in time.

5. Logistics — the supply and resupply of people, things, and services that are necessary for sustained operations.

You will note that communications does not occur in this basic set of functions because one does not communicate for the sake of communicating. Communications is an organic part of each of the five basic functions described. Nevertheless, communications — because of both history and customary operating procedures — usually is considered as the sixth basic function.

These functions are essential to all levels of complexity. They apply to the simplest warfare system — a man in combat without external aids — and equally as well to a very complex system, such as the Atlantic Fleet.

There are many levels of complexity, within the military, that can be described. However, three general levels of complexity of man-machine systems appear to be adequate for our purpose; in the Navy these can be described as:

Force Systems — for example, a Hunter Killer Group, a Carrier Attack Force, a Marine Combat Team, or the numbered fleets;

Force unit systems or vehicle systems — such as a destroyer, a carrier, a cruiser, an amphibious vehicle, or a tank, each considered as a complete, combat-ready entity; and

Primary systems — such as a TALOS weapon system, a destroyer communication system, or an integrated shipboard ASW surveillance system. Primary systems are subsystems of a Force Unit System. In general, they represent the lowest complexity level considered in long range planning.

Within this spectrum of complexity, the vehicle systems appear to be those upon which we should center our attention at this symposium.

It is upon our vehicle systems that the force of enemy weapons — blast, shock, vibration, radiation, and so on — is exerted. It is upon our vehicle systems that the reaction of our own weapons — shock, vibration, overpressure, and thermal shock — is exerted. It is upon our vehicle systems that the reaction of natural environment — wave impact, aerodynamic heating, pressure, and so on — is exerted.

My primary thesis, then, will be personal observations on trends in military vehicle systems. I must emphasize that these remarks are my own prognostications and have no official status. They are only possibilities gleaned from reading about the present and thinking about the future.

Our entrance into the space age has forced us to consider the earth as an entity — a unity. The earth sciences have become more important to us from both military and civilian points of view.

Geodesy, meteorology, and oceanography are vital to our overall consideration of the environment in which we are immersed. The importance of geodesy is readily observed from the open literature. Not so obvious, however, is the fact that prediction techniques for accurate 30-day forecasts, as well as 5-day and 1-day forecasts, of both weather and oceanographic parameters must be attained. Even beyond this, it becomes evident that we must strive toward some degree of capability for controlling our environment both in the atmosphere and in the ocean.

There are other observations that stem directly from this consideration of the earth as a unity. For instance, the use of submerged devices and submarines has, I believe, become several times more important than before. Both ships and submarines must become as independent of port facilities as possible. Land vehicles must be made independent of road systems insofar as this is technically possible. Air transport must also change. Higher speeds, greater capacity, and longer range seem important. Also, aircraft must become independent of long, prepared runways.

Hence combination types of vehicles, such as hydrofoil craft, ground effects machines, and V/STOL aircraft, appear to be assuming roles of increasing importance. These are the trends which, I suggest, will be most important in setting the "vector direction" along which you must plot your long-range programs for studying environmental effects. I further suggest that your most difficult problems are to determine what will be the critical parameters of the environment. Once these have been established, it

should be possible to see clearly what is involved in design for compatibility with the environment. In order to do this, I suggest that we must understand fully the sea, the air, and the space within which we live.

I am tempted, I confess, to stop at this point because all of my predictions have been given. Nevertheless, I indicated that I would give you one man's guess on technological projections for the next decade.

Since my current habitat is the Navy, allow me to start with inner space — the ocean — and progress to outer space in terms of what I believe to be technically feasible during that time period. In order to be brief, I shall consider, in each case, only the one parameter that will act as the "pacing" parameter.

Within the next decade, explorations of the deepest ocean trenches will be routine. The first expeditions of this type have, as you know, already been accomplished by the Navy Electronics Laboratory using the bathyscaph Trieste. Although there are numerous problems in instrumentation and operations associated with deep-ocean research vehicles, only one environmental parameter assumes importance — pressure. Water tight integrity is an obvious necessity. No new problems involving vibration, shock, and so on, are anticipated.

By 1975 it could be possible for submarines to remain on station submerged for 90 days or more, at depths of 4000 feet with sustained speeds in excess of 35 knots. Boundary layer control may make possible much lower resistance for a given hull shape. As in the case of the deep ocean research vehicles, pressure is the only environmental parameter that will change appreciably. The major problem, of course, is the development of new materials and structures to withstand these pressures. In addition, this one change places a premium upon the effectiveness with which associated problems are solved. For instance, water tight integrity cannot be endangered by excessive vibration. Submarine weapons must react quickly and silently. It is possible that mounting weapons external to the pressure hull will provide an attractive solution. In any event, efficient seals, rotating joints, bearings, and vibration and shock isolation will be required.

During this same period surface ships might be capable of sustained speeds in excess of 35 knots in high sea states. Such progress would depend, of course, upon major gains in drag reduction of hulls and improved propulsion systems. Drag reduction of as much as 25 percent might be realized. Research on supercavitating and superventilating propellers could provide the possibility for appreciable increases in speed.

A naval ship is primarily a mobile platform for carrying a payload of people, electronics, and weapons. In combat this combination must work effectively, in concert, both to destroy the enemy and to nullify his attack. Hence, many serious problems can be expected to accompany anticipated increases in speed.

For instance, structural flexure of the ship could introduce a significant degradation into the overall accuracy of inertially guided weapon systems. Methods for compensating for structural flexure may have to be developed.

Increased speeds can be expected to lead to greater use of automatic and centralized controls. These are subject to derangement by shock, vibration, radiation, temperature, and so on. Although these are not new problems, their solution will be crucial to effective operation. In essence, this problem embraces the total subject of reliability/maintainability of electronic systems, feed-back control systems, and so on.

A definite trend can be observed in the use of men as components of these systems: men will be used less and less as a source of motive power, and more and more as decision-makers and nonlinear, feed-back, control devices. This trend will force all of you to think long and hard about how to provide an optimum environment for the men. The problems introduced are extremely complex. For instance, to reduce the probability of physical injury to a man one should reduce, as nearly to zero as possible the relative motion of the man with respect to his immediate surroundings. This same condition pertains for effective operations involving direct man-machine interactions. In those operations that involve only the man, accelerations must be reduced as nearly to zero as possible in order to maximize the man's effectiveness. You will agree, I am sure, that attaining both of these conditions, simultaneously, in all manned spaces of a ship presents some serious problems. And yet, these conditions are not far-fetched, because it appears that they are equally applicable to most classes of electronic equipments.

During the time period under consideration, defense against bacteriological and chemical munitions may become essential. The most probable solution to this problem will be to hermetically seal all manned spaces and maintain them at a slight positive pressure of clean,

filtered air. A variety of subsidiary problems, such as temperature and humidity control, vibration and shock proof seals, and so on, may be expected to be encountered as a result of this condition.

Hydrofoil craft in the 500-ton class, operating at speeds as high as 90 knots, could possibly be built. The new environmental problems that will be encountered in connection with hydrofoil craft also are those associated with high speeds in high sea states. One can anticipate that the seriousness of shock, vibration, and related problems will increase at least linearly with speed, and perhaps as rapidly as the square of the speed.

Ground effects machines, capable of effective operation over both land and water, are possible. It is conceivable that these craft, powered by gas turbines, could attain operating speeds of 100 knots or more. design for unexpected encounter with obstacles or for power failure will be necessary. Hence, major problems in shock abatement will be encountered.

In the next decade the development of military amphibious and ground vehicles will probably involve improvements of known techniques. Major gains can be anticipated in off-road speed and mobility, reduced weight, and increased range. A significant advance may be attained through the development of multi-fuel, compression-ignition engines having reduced dimensions, weight, and specific fuel consumption. The general trend will be toward air-portable and air-droppable vehicles.

In this case also, major problems in temperature and humidity control and shock abatement may be introduced by the need for such vehicles to be impervious to attack by CW/BW munitions.

One can also anticipate the routine use of low-flying, high-speed aircraft that follow the contour of the earth. There will be, I suspect, major structural problems induced by vibration and buffeting as well as aerodynamic heating to contend with.

Speaking generally, one can predict aerospace craft operating from sea-level to 300,000-foot altitude and at speeds ranging from 0 to 25,000 ft/sec. VTOL aircraft may be expected to operate over the speed and altitude ranges that pertain today for conventional aircraft. Many problems will be encountered in the design of these families of vehicles. For instance, stagnation temperatures as high as 2500°F could be encountered on leading edges. In all probability, ionized boundary layers would accompany the aerodynamic heating and introduce a host of new problems. Internal temperatures in ducting and engine enclosures might rise as high as 3000° to 4000°F. Rain erosion could be expected to be a serious problem. The combined environment of dynamic loading, thermal stress, acoustic loading, and ionization could be expected to introduce major problems in fatigue behavior as well as structural design. At present, these extreme conditions are encountered for only a brief period of time. That may not be true in the future.

At extreme altitudes, of course, many new aspects of environment will become important. At low ambient pressures, heat discharge from the vehicle will be reduced greatly: Solar and Cosmic radiation and so on, will be encountered. The current literature is replete with the environmental problems of satellites and space travelers. I shall, however, add one more point: When the first man arrives on the moon, his safety will be completely dependent upon some set of shock absorbing devices. This, of course, brings us right back to the class of problems that led to the formation of this symposium a decade ago.

* * *

Section 2

DEVELOPMENT OF SPECIFICATION REQUIREMENTS

DERIVATION OF SHOCK AND VIBRATION TESTS BASED ON MEASURED ENVIRONMENTS

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The numerous considerations pertinent to simulating the damaging effects of shock and vibration environments are presented and discussed. The purpose of this progress report was to stimulate discussion on a controversial subject. It was hoped that the S2-X-46 Committee might learn of current significant work or opinions which have a bearing on the project and which could be incorporated as influencing factors in the Committee's final report at the end of the year.

INTRODUCTION

The American Standards Association Section Committee S2 on Shock and Vibration has organized an exploratory group designated as S2-X-46 with a 1-year term to examine the subject of "Testing to Simulate the Shock and Vibration Conditions Encountered in Actual Environments." The objective set for S2-X-46 is to determine the feasibility of preparing a standard practices document to represent the present state-of-the-art on the subject of specifying laboratory simulation of shock and vibration environments.

The material presented here is in the nature of a progress report and represents a brief summary of the subject matter which has been reviewed by the S2-X-46 Exploratory Group. The objective of this report is to stimulate discussion on a controversial subject so that the S2-X-46 Exploratory Group may learn of current significant work which might otherwise be overlooked and which may have a bearing on its feasibility report to the American Standards Association. Engineers having data pertinent to environmental shock and vibration simulation are requested to report such information to the Chairman of S2-X-46 either directly or through individual members on the committee.

The members of S2-X-46 have been selected to represent a wide cross section of shock and vibration personnel in government and industry, and also a wide diversity of opinion on the subject of laboratory shock and vibration simulation. The members of S2-X-46 are listed in the Appendix.

S2-X-46 APPROACH

The approach taken by S2-X-46 has been to consider the subject of shock and vibration simulation in its broadest sense. Two diametrically opposite extremes in approaches to shock and vibration simulation have been pursued for purposes of the present investigation. These are (1) exact duplication of environment, and (2) simulation of the damaging characteristics embodied in the environment. The concepts involved are discussed in the following sections.

EXACT DUPLICATION

In concept, exact duplication of an environment is simple and direct. If the laboratory

environment is an exact reproduction of the real environment, then it must certainly possess the same damaging characteristics as the real environment. This approach makes it unnecessary to know a priori which features of the environment cause damage in equipment. In this sense, exact duplication is considered to be an ideal approach for laboratory testing. In the laboratory, however, aside from the technological problems of exact reproduction of environments, there are certain basic and practical questions which must be considered. They are as follows:

1. Exact duplication implies a unity scale factor between the real environment and testing time. If the service environment is long, then an unreasonable amount of testing time may be called for in the laboratory.

2. If an equipment may be used alternatively in several different environments, it becomes impractical to reproduce each environment sequentially in the laboratory.

3. Environment severity in vehicles tends to be statistical in nature. Hence, it becomes a question in reliability statistics to determine which particular sample environment should be selected for exact reproduction in the laboratory.

4. Exact reproduction of environment in the laboratory implies that the effect of the equipment interaction on the testing machine, i.e., impedance match, will be the same as in actual service. In the present state-of-the-art there is no mechanical impedance information on service installations to permit impedance duplication under laboratory conditions.

For the reasons noted, S2-X-46 believes that exact reproduction of environment in the laboratory is a highly specialized research technique with but limited application to general laboratory use. Opinions and experience, pro or con, among reviewers of this report are solicited for consideration by S2-X-46.

SIMULATION OF DAMAGING EFFECTS

The alternative to striving for exact duplication of environments in the laboratory is to synthesize a test which will produce the same damage as the real environment. This approach requires a knowledge of the mechanism of failure in equipment when subjected to dynamic inputs. Unfortunately, mechanical failure and malfunction of equipment as a result of shock and vibration is a highly complex subject and there is very little published experimental information. There are, however, some data (Refs. 1 and 2) on vibration tests to failure of simple electronic components, and from these we can deduce some of the basic considerations involved in simulating equipment failure conditions in the laboratory.

Figure 1 shows the results of resonant vibration tests to failure performed on several sizes of capacitors and resistors which were soldered to terminals by standard techniques. The plotted points represent the vibratory acceleration amplitude $\ddot{u}$ applied to the base of each component and the number of applied cycles N at which failure resulted. It can be observed that the characteristic slope of the curves drawn through families of failure points is like that of fatigue (σ-N) curves for materials.

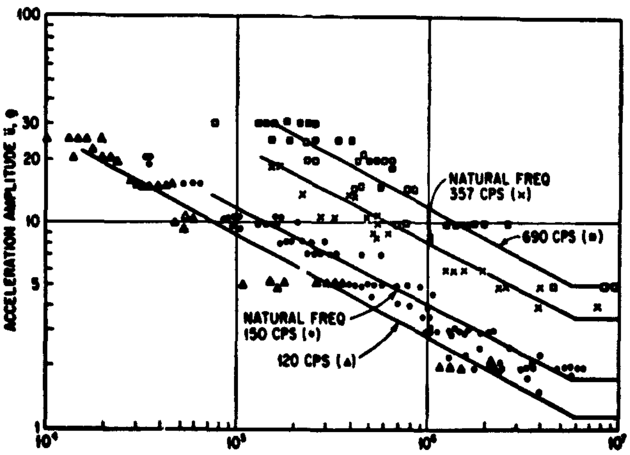

Fig. 1 - Vibration tests of electronic components

The $\ddot{u}$ - N failure curves even suggest an endurance limit "knee" similar to that of conventional σ - N fatigue data for materials.

Failure Surface Concept

The data in Fig. 1 are plotted for two parameters, $\ddot{u}$ versus N, with a third parameter specified as the resonant frequency of the component. If samples of each component had been tested at off-resonant frequencies, curves similar to those in Fig. 1 would have been obtained, except that the applied vibratory accelerations would have been proportionally higher, depending on damping (maximum transmissibility or Q) and proximity to resonance.

With experimental data of the type shown in Fig. 1 for both resonant and off-resonant conditions, it would be possible to define a three-dimensional "failure surface" for particular components and items of equipment. The failure surface parameters are input vibratory acceleration $\ddot{u}$, vibration frequency f, and number of cycles to failure N. Figure 2 illustrates a hypothetical failure surface for an equipment having one resonant frequency. The resonance appears as a valley in the failure surface because, depending on the amount of damping present, only a small input is required to produce failure. The failure surface for a complex equipment having several natural frequencies would exhibit a valley corresponding to each natural frequency.

If a failure surface such as that shown in Fig. 2 is presumed to exist for every equipment, it is possible to hypothesize certain requirements for a laboratory test. In Fig. 2, line "a" defines an environment frequency — amplitude spectrum for a given steady-state vibration condition which is presumed to be present at all times. Since the environment exists for the same amount of time at low frequencies as at high frequencies, the line "a" appears as a diagonal slice increasing to higher values of N as frequency f increases. If the equipment failure surface has a well defined endurance limit "knee," then it is unnecessary to test beyond N, the number of cycles corresponding to endurance limit stress in the equipment. The environment spectrum crosses through the valley in the failure surface at frequency band AA' thus indicating that failure will occur when vibration is applied at or near the resonant frequency for a number of cycles corresponding to N_{el}. This failure would be detected in the

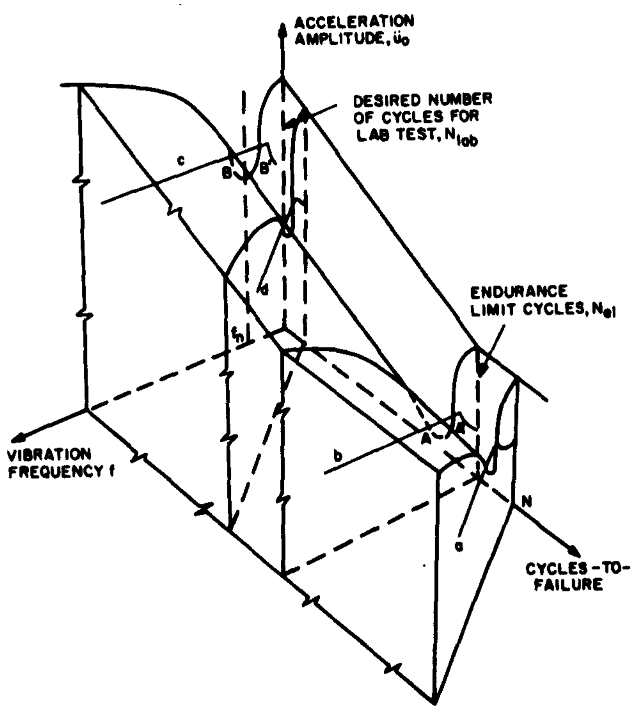

Fig. 2 - Concept of laboratory test based on failure surface

laboratory after an appropriately long test at or near the resonant frequency. The laboratory test could be shortened appreciably if the test amplitude were increased in some manner related to the slope of the $\ddot{u}$ - N curve of the resonant valley in the failure surface. This is indicated by line "c" which indicates that failure will occur across the frequency band BB' for a short duration test based upon applying a number of cycles corresponding to N_{lab}.

Synthesis of Equipment Failure Curve

In the present state-of-the-art there are insufficient experimental data to establish failure surfaces for equipment which could be used as damage criteria for developing laboratory testing procedures. It appears possible, however, on an interim basis to utilize the published data on internal damping and endurance properties of materials as a means for synthesizing resonant $\ddot{u}$ - N curves. This approach is indicated in Fig. 3. The construction in Fig. 3 is based upon a derived relationship between acceleration $\ddot{u}$ and stress, in Ref. 1, as follows:

$$\frac{Q\ddot{u}}{Q\ddot{u}_{e\ell}} = \frac{\ddot{u}}{\ddot{u}_{e\ell}} = \left(\frac{\sigma}{\sigma_{e\ell}}\right)^{1.4} \tag{1}$$

where the subscript "$e\ell$" denotes the value of support acceleration $\ddot{u}$ or response stress corresponding to the endurance limit of the material. The development of Eq. (1) utilizes experimental data in Ref. 3 on the internal damping energy dissipated per cycle of vibration for several structural materials. For purposes of simplifying the construction of the failure curve in Fig. 3, the acceleration corresponding to endurance limit stress is normalized as unity and is plotted corresponding to $N_{e\ell} = 5 \times 10^6$ cycles. This is a representative value of endurance limit cycles which frequently appears in the technical literature on endurance properties for a wide variety of materials. Similarly, based on a further generalization of published data, the applied vibratory acceleration corresponding to twice the endurance limit stress is plotted at $N = 10^4$ cycles. The resonant $\ddot{u}$ - N failure curve is then constructed as a straight line on log log coordinates between the two aforementioned points. Note that the synthesized curve is drawn as a straight line without a knee at the endurance limit acceleration. The justification for extending the failure curve as a simple straight line for all values of N is that this conservatively predicts failure sooner than might actually occur in practice. Moreover, this permits the derivation of a simple equation defining the failure curve, as follows:

$$N = \frac{5 \times 10^6}{(\ddot{u}/\ddot{u}_{e\ell})^{6.5}}. \tag{2}$$

Concept of Cumulative Damage

The limitation of the failure curve synthesized in Fig. 3 is that it is based on a constant

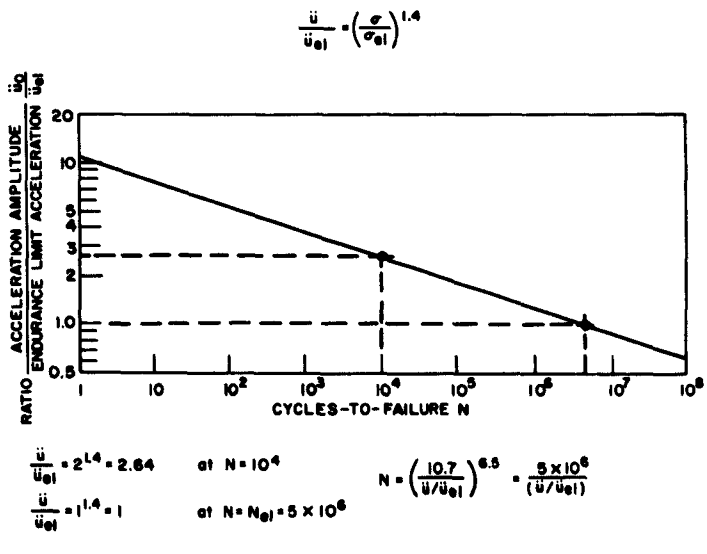

Fig. 3 - Construction of $\ddot{u}$-N failure curve for vibration at resonance

amplitude vibratory condition existing until failure occurs. In both real environments and in certain laboratory environments the input amplitude will vary from cycle to cycle. It is thus necessary to adapt Fig. 3 to a criterion of failure wherein the damage contribution by each of several varying amplitudes can be assessed. There are many concepts of fatigue damage accumulation under varying amplitude conditions. References 4, 5, and 6 present three such concepts with varying degrees of analytical sophistication introduced to minimize differences between calculations and experimental results. The simplest useful hypothesis is that of Ref. 4 and is depicted in Fig. 4. In the approach illustrated in Fig. 4, failure is considered to occur when the summation of the ratios of applied cycles n to failure cycles N for each amplitude is equal to a constant. This is expressed mathematically as:

$$\sum \frac{n_i}{N_i} = C. \tag{3}$$

determine whether failure will be incurred in equipment when subjected to field measured vibration amplitudes for a time duration related to service life. Equipment which will be in continuous service for a long period of time, say several months or more, should be laboratory vibration tested for a period of time which will result in applying the endurance limit cycles N_{el} as depicted in Fig. 2. Alternatively, the testing time may be reduced by increasing the test amplitude in accordance with Eq. 2 or Fig. 3.

Ideally, the vibration levels measured in service should be applied only at the resonant frequencies of the equipment undergoing test. In practice, the equipment resonance is not known precisely, consequently, the test is performed by varying the frequency in small increments until the entire frequency band is covered. In the limit, as the frequency increments become very small, the test becomes a sweep-frequency test. Therefore, the rate of change of test frequency, or sweep speed, becomes important in order to insure that the system undergoing test has time to respond with maximum resonant amplification.

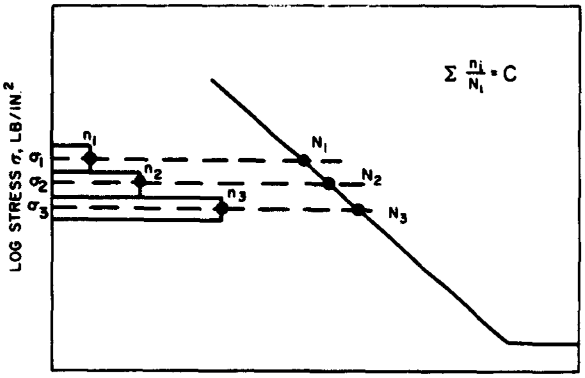

Fig. 4 - Concept of cumulative fatigue damage

In various fatigue experiments which have been performed with stress amplitudes which vary randomly or involve cycling between small and large amplitudes, the constant C has been found to vary between 0.2 and 0.8, see Refs. 7 and 8. This suggests the use of an average constant $C = 0.5$ for interim purposes in deriving test procedures.

Suggested Application of Failure Criteria in Derivation of Sweep Vibration Test

In concept a laboratory vibration test is a resonant fatigue test. The objective is to determine whether failure will be incurred in equipment when subjected to field measured vibration amplitudes for a time duration related to service life.

The effect of "sweeping" the test frequency f through a resonant frequency f_n at various sweep speeds β is discussed in Refs. 1 and 9. This effect is illustrated in Fig. 5 for a single degree of freedom system whose maximum transmissibility or $Q = 20$. Figure 5 presents envelopes of system maximum responses as a function of test frequency and a parameter N_n, which is related to sweep speed as follows:

$$\beta = df/dt = f_n^2/N_n. \tag{4}$$

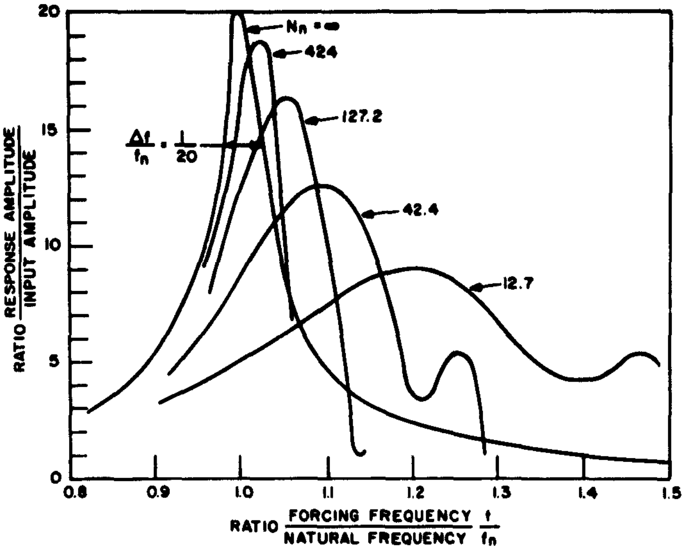

Fig. 5 - Response to sweep frequency

The parameter N_n is also related to the number of cycles ΔN which will be applied during a sweep through a frequency band Δf, as follows:

$$\Delta N = \frac{\Delta f}{f_n} N_n \,. \tag{5}$$

or, alternatively

$$\Delta N = \frac{1}{Q} N_n \,. \tag{5a}$$

It may be seen in Fig. 5 that a value of N_n greater than approximately 300 is necessary for a system with low damping to approach at least 95 percent of its theoretical steady-state response at resonance. It can also be seen that significantly high stress cycles occur in the frequency band defined by $\Delta f/f_n$ as the test frequency approaches and recedes from the resonant frequency. These cycles are less than the maximum at resonance, but are still large enough to inflict fatigue damage on the structure. This effect can be evaluated by (1) summing the cycle ratios corresponding to various levels of response as the test frequency sweeps through resonance, (2) dividing the response level by Q to obtain an equivalent steady-state vibration input level, and (3) substituting the foregoing in Eqs. (2) and (3) with $C = 1/2$, as follows:

$$\sum \frac{n_i}{N_i} = \frac{1}{2}$$

$$\sum n_i \times \frac{(\ddot{u}_i/\ddot{u}_{e\ell})^{6.5}}{5 \times 10^6} = \frac{1}{2}$$

$$\sum n_i \, (\ddot{u}_i/\ddot{u}_{e\ell})^{6.5} = 2.5 \times 10^6 \,. \tag{6}$$

Denoting the input at resonance by $\ddot{u}_{max}$ and rewriting the off-resonant equivalent inputs as ratios of $\ddot{u}_{max}$, Eq. (5) becomes:

$$\sum n_i \left(\frac{\ddot{u}_{max}}{\ddot{u}_{e\ell}}\right)^{6.5} \left(\frac{\ddot{u}_i}{\ddot{u}_{e\ell}}\right)^{6.5} = 2.5 \times 10^6 \,. \tag{7}$$

A numerical evaluation of Eq. (7) in Ref. 1 for a sweep through a frequency band Δf where the response is greater than the rms value, i.e., greater than 0.707 of maximum response, produces the following result:

$$\Delta N \left(\frac{\ddot{u}_{max}}{\ddot{u}_{e\ell}}\right)^{6.5} = 5 \times 10^6 \,. \tag{8}$$

Equation (8) thus defines the condition for cumulative fatigue failure during cycling through a resonance. Now, to relate this to a failure which is due to a single frequency test directly at resonance, combine Eqs. (2) and (3) with $C = 1$, as follows:

$$\sum \frac{n}{N} = \sum n \times \frac{(\ddot{u}_{max}/\ddot{u}_{e\ell})^{6.5}}{5 \times 10^6} = 1 \tag{9}$$

(Cont.)

$$\sum n\left(\frac{\ddot{u}_{max}}{\ddot{u}_{e\ell}}\right)^{6.5} = 5 \times 10^6 \qquad (9)$$

Comparison of Eqs. (8) and (9) shows that, to cause the same damage when the acceleration amplitude of the support $\ddot{u}_{max}$ is the same, the number of cycles of vibration ΔN in a sweep frequency test is identical to the corresponding number Σn in a constant frequency test.

The use of Eq. (8) can be seen upon combining Eqs. (4) and (5a) and writing the expression for sweep rate as follows:

$$\beta_{\Delta N} = \frac{df}{dt} = \left(\frac{1}{\Delta N \times Q}\right) f_n^2 \text{ cps/sec.} \qquad (10)$$

Equation (10) is plotted in Fig. 6.

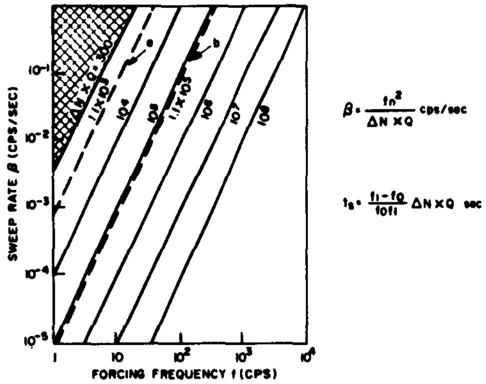

Fig. 6 - Sweep-frequency test with constant N

In Eq. (10), ΔN may be regarded as the number of significant cycles which will be applied at each resonant frequency of a given Q (maximum resonant transmissibility) during one sweep from minimum to maximum frequency. The number of sweeps is determined by the total number of cycles which may be desired. An equation for the time duration $t_{\Delta N}$ of one sweep with constant ΔN in the response can be determined by separating the variables in Eq. (10) and integrating over a range of frequencies from f_0 to f_1. This results in the following equation:

$$t_{\Delta N} = \frac{f_1 - f_0}{f_0 f_1} \Delta N \times Q \text{ sec.} \qquad (11)$$

If the test is to be performed on equipment designed for long service, as in aircraft, then the number of applied cycles should be $\Delta N = 5 \times 10^6$ with a test amplitude that is representative of the actual environment. If this test duration is too long to be convenient, it may be reduced by selecting an appropriate test amplitude exaggeration factor from the failure curve of Fig. 3.

When the anticipated environment is short, as is the case for missiles and rockets, the sweep cycling procedure should preferably be based upon applying constant increments of time Δt at the rms response bandwidth Δf. The sweep speed equation can then be written as

$$\beta_{\Delta t} = \frac{df}{dt} = \left(\frac{f_n}{\Delta N \times Q}\right) f_n = \left(\frac{1}{\Delta t \times Q}\right) f_n. \qquad (12)$$

Equation (12) is plotted in Fig. 7. The time duration $t_{\Delta t}$ to make one frequency sweep with constant Δt can be determined by separating the variables in Eq. (12) and integrating over a range of frequencies from f_0 to f_1. This produces the following equation:

$$t_{\Delta t} = Q \times \Delta t \log_e \frac{f_1}{f_0} \text{ sec.} \qquad (13)$$

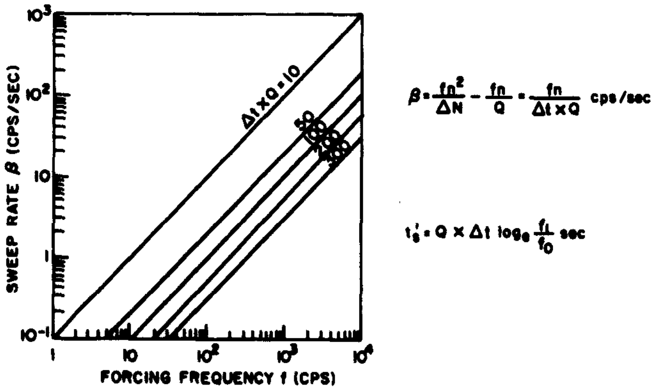

Fig. 7 - Sweep-frequency vibration test with constant increments of time

The time increment Δt should be the same as the time duration of the real environment for a test amplitude that is representative of the real environment. If the resulting sweep test duration $t_{\Delta t}$ is too long for convenience in the laboratory, then a test amplitude exaggeration factor can be selected from Fig. 3 for any desired percentage reduction in test time. Conversely, the data in Fig. 3 could be applied to increase the test duration by selecting an appropriate reduction for the test amplitude.

SUMMARY

This progress report of the American Standards Association S2-X-46 Exploratory Group has presented and reviewed many of the factors relative to "exact duplication of environments" in the laboratory as opposed to "simulation of damaging effects." It is pointed out that exact duplication is conceptually an ideal test, but there are too many practical shortcomings for its use other than as a research tool. Simulation of the damaging effects of environments in the laboratory is the alternative to exact duplication.

The approach involving simulation of damaging effects has been successfully applied in the past, generally in combination with empirical cut-and-try methods. The empirical aspects of the "simulation" approach can be minimized by establishing and utilizing criteria for equipment failure, or malfunction, based on simple vibration tests to failure at different levels of input. A sample approach to the development of a sinusoidal sweep frequency test is presented based upon a synthesized $\ddot{u}$ - N (vibratory acceleration input-versus-cycles to failure) curve for equipment. The synthesized $\ddot{u}$-N curve is employed in conjunction with a concept of cumulative damage under varying amplitude input conditions. These then represent the necessary failure criteria which are essential for developing all shock and vibration (sinusoidal or random) laboratory test procedures.

Comments and data on current significant analytical or experimental work which may resolve the question on the feasibility of the approaches and concepts reviewed herein, are solicited from industry and government by the S2-X-46 Group.

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Appendix

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ENVIRONMENTAL TESTING STANDARDIZATION VIA MIL-STD-810 ENVIRONMENTAL TEST METHODS FOR AEROSPACE AND GROUND EQUIPMENT

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An urgent need for the upgrading and standardization of test procedures and criteria in the Air Force has been met by the development of MIL-STD-810. This paper explains the purpose of the standard, what it is and what it is not, and the way in which it was developed. The test methods of this standard apply broadly to all items of aerospace and ground equipment except airframes and primary power plants.

INTRODUCTION

MIL-STD-810 (USAF) has been prepared to bring together in one document procedures and guidance for the environmental testing of future generations of aerospace and ground equipment.

The general provisions of MIL-STD-810 include standard laboratory test methods for determining the environmental suitability of military hardware when used under similar service conditions, guidance to the system or equipment engineer formulating an environmental test program, and guidance for those preparing the environmental test portions of detail specifications. Contained in the standard are 18 basic test methods with a total of 26 detailed test procedures. Excluded are tests for airframe structures and primary power plants. Tests required for these items, as well as design factors for broad coverage areas such as the suppression of radio frequency interference, effects of corrosive fuels and oxidizers, and so on, which in a general sense constitute environments, are either adequately covered by existing specifications or, by their nature and complexity, must be treated separately.

The environmental stress levels stated in the test methods of MIL-STD-810 represent what is generally considered to be the extreme conditions which usually constitute the minimum acceptable conditions for world wide military use. However, when designing to specific requirements where it is known that the environmental stress conditions are more or less severe than those given in the standard, the test limits may be adjusted as necessary. These, and other such details, must be determined by the engineer and included in the detail specification. It is not intended nor should it be expected that this document, prepared to cover the broad field of environmental testing, be a substitute for the engineer's common judgment and decision making responsibility.

In referring to specifications, it is pointed out that MIL-STD-810 is not a specification nor is it intended to be. The contents of this document do not constitute the necessary ingredients for a specification. By military description a specification is the governing document for an item of hardware. It states the minimum acceptable requirements to insure that the item will do exactly what it is intended to do along with those acceptance tests and quality assurance provisions peculiar to the item. Since MIL-STD-810 is not applicable to any one specific item, or even a class of closely related items, it can not be considered a specification.

It is recognized, with no apology intended, that MIL-STD-810 in its present form is not a panacea for every problem associated with environmental testing. The contents of the test methods are not based on any newly performed basic research program or revolutionary findings which would challenge the state of the art.

It does bring together, in one document, standard test methods that are at least current with the state of the art. For those who have not reviewed this document, or for those not acquainted with its origin and underlying philosophy, a brief history and chronology of events may be of interest.

BRIEF HISTORY

Over the past several years numerous complaints have been heard regarding the environmental test procedures contained in various military specifications. These complaints have ranged from comments that the military has failed to lead the state of the art in environmental criteria and testing, to charges of redundancy, conflict, and inconsistencies in the performance of tests among various specifications. The problem is traceable, in part, to the gradual evolution of the system concept as implemented by the Air Force; it was more noticeably observed, however, as system contractors began integrating various equipments, tested against different specifications, into highly complex flight vehicles each with its own peculiar environmental profile. The growing problem of adequate environmental testing techniques and facilities was further hastened by breakthroughs in rocketry and the race for supremacy in space. Deficiencies in the overall environmental program, when viewed from the system level, became more and more apparent.

A study was subsequently initiated to determine what measures could be taken to effect the standardization of environmental testing to support the system concept. Attention was focused on the following five specifications: MIL-T-5422, Environmental Testing for Aircraft Electronic Equipment; MIL-E-5272, Environmental Testing, Aeronautical and Associated equipment; MIL-E-4970, Environmental Testing, Ground Support Equipment; MIL-A-26669, Acoustical Noise Tests for Aeronautical and Associated Equipment; and one remaining document, MIL-S-27507, Shock Test, Saw Tooth Pulse, which was about to be published. The result of this study clearly justified the need for upgrading test procedures, criteria, and the requirement for one standard environmental test document. The decision was made to continue the effort by preparing a completely new environmental test document.

PREPARATION OF FIRST DRAFT

In preparing the general framework for the first draft of the standard it was decided that only general guidance, philosophy, and other criteria common to the majority of test methods would be placed in the basic portion of the document; that type of information which will probably see little or no change. Expecting that some tests would require frequent change to keep pace with the state of the art, each test was assembled as a separate "method." This arrangement permits revisions to any particular test method without disrupting the entire document.

Having established the format, the organization of the individual test methods was next approached. The previously mentioned documents, which may be called "donor" specifications were reassembled by environment as "raw data" and critically analyzed. This "weeding out process" not only revealed discrepancies in and among the various tests for the same environment or condition, but also brought to light certain methods and tests which were still geared to pre-World War II technology. In this analysis the "raw data" were required to satisfy the following questions: Were the environment and the purpose for conducting the test adequately described? Were specialized apparatus, control of environmental conditions, and handling and mounting problems peculiar to the test item clearly defined? Was the test itself presented in a clear and logical manner? Were the technical parameters of the test "scientific" in the sense that the test engineer could rely on the results obtained? In most instances the raw data failed this test. The development of each test method was approached with caution and some apprehension regarding extensive changes to the technical organization and stress levels of the tests. It was assumed that the "raw data" derived from the "donor specifications" were essentially correct. This assumption eventually proved false in many cases.

REVISION OF FIRST DRAFT

After some months the first draft of the standard was ready for coordination. It was decided to circulate the draft as widely as possible along with a request for comments and suggestions. A sincere engineering evaluation was wanted from the military and from the aircraft, aerospace, and electronic industries as well as independent testing laboratories. A large response was not anticipated to this invitation for comments. Again, this complacency was engendered by the assumption that the "donor data" were correct. Over 200 copies of the draft were circulated. A most rewarding response of 111 replies was received. The majority of these replies, particularly from

industry, urged the continuation of the effort for one standard environmental document. A number of activities, both military and industrial, provided as many as four or five pages of detailed technical comment. These comments predominantly pointed out engineering deficiencies and other weaknesses in the test procedures, the data for which, almost without exception, were inherited from the "donor specifications.

As was expected, some replies were received which challenged or repudiated the basic intent of the effort. It was suggested that better use be made of man-hours by revising the "donor" specifications and that the introduction of still another environmental document would only add to the general confusion. These allegations are answered as follows: Patching up the old specifications was not considered realistic or expedient. Related as they are to specific items, i.e., aircraft electronic and ground support equipment, it was believed that any attempt to introduce into them advanced requirements at the overall systems level would have posed serious problems. Further, the procedures involved in accomplishing routine changes for just one specification are most time consuming. Considering the number of specifications involved and the resolution of many technical differences the attempt to "catch up" would have been perpetuated infinitely.

The months that followed were devoted to setting things in order. Based on the comments received, and on in-house engineering, each test was again analyzed, re-engineered, and re-edited many times. Not one test derived from the "donor" specifications remained completely unchanged. Some test methods underwent extensive engineering effort. Typical of these are tests in the dynamics area, especially vibration and acceleration. The details of each individual test method will not be discussed here, but it is worthy to note that through the joint effort of dynamics engineers, both in industry and in the military, the dynamics portions of MIL-STD-810 reflect significant and improved technology. However, it is now obvious that additional progress still can and must be made in this area. Notwithstanding, this overall cooperative effort clearly illustrates what can and has been done when military and industrial environmental engineers are afforded the opportunity to communicate informally and resolve mutual problems.

MIL-STD-810 (USAF)

The result of this effort is reflected in MIL-STD-810 in its present form. Those who may feel concern or alarm that MIL-STD-810 tests may not now suit their particular requirements are invited to compare a test from the new standard with a like test from one of the "donor" specifications. It will be found that the MIL-STD-810 tests provide more positive guidance and test technology than heretofore.

MIL-STD-810 should not be applied in retrospect. It is fully realized that the "donor" specifications are listed as applicable documents in many military procurement specifications. The intended application of MIL-STD-810 is for environmental testing related to new systems engineering and design. It may be assumed, however, that as MIL-STD-810 is more widely applied, these other specifications will gradually fall into disuse. Together the Military and industry have resolved many environmental problems in the development of this document. However, when considering the broad scope of this effort, some differences of opinion are bound to prevail. Some controversial points may never be resolved.

In an organization such as the Aeronautical Systems Division with its many systems project offices and their responsible system development contractors, environmental problems arise almost daily, especially in the initial phase when the system environmental compatibility for a new flight vehicle is being developed. It is essential that constant surveillance and liaison across this vast military-contractor effort be maintained so that MIL-STD-810 can be kept up to date. If, through the use of this standard, deficiencies are noted or if new criteria and methodology are developed for which there is a general need, it is suggested that recommendations be made for upgrading the standard. The importance of this team effort can not be minimized.

Specifically, procedures for complete environmental testing of the requirements for space vehicles, and meaningful test methods for combined environmental testing should be introduced into this document as quickly as the technology can be developed.

Confer with, or write to Aeronautical Systems Division, W-PAFB, Ohio, Attn: ASTEVC, regarding changes or revisions to this standard.

DISCUSSION

B. Wigle (Martin Orlando): I would like to compliment you on your fine document. I think it something long needed. I would like to question the vibration curves and where the multiple cross over points came from and how we should perform these tests?

Mr. Golueke (ASD, Chairman): I have had a lot of input to that particular document and your particular question. First I want to say that Mr. Junker is responsible for the overall document and obtains all his inputs from the engineering talents back at Wright Field. Now with regard to your question — what has been done, is that recently we've had some dynamics conferences with industry at the Field and we are already considering changing those cross over points — mainly towards the low-frequency area.

D. Stern (GE): With reference to the shock portion, here it would seem from the way the spec is written that the sand pounders are out? Is this true?

Mr. Junker: That is quite correct.

Mr. Golueke: I might answer that question too. Yes, the sand pits are out and in relation to this particular document it was interesting to note at the conference I mentioned that we did take what was called an opinion vote of the conferees in relation to shock. Four techniques were involved. Do you want to call out shock requirements by means of the spectra? Do you want to call them out by means of the time history, the pulse shape, a combination of the two, or do you want a machine? It was interesting to note that the majority wanted the pulse shape and not necessarily the spectrum. They knew what the spectrum would be, but the document as it is, calls out the pulse shape and it is quite possible that it will remain as such.

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DEVELOPMENT OF MILITARY SPECIFICATION MIL-T-23103 (WEP), GENERAL REQUIREMENTS FOR THE THERMAL PERFORMANCE EVALUATION OF AIRBORNE MILITARY ELECTRONICS EQUIPMENT

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A program was set up to define valid and useful measures of electronic equipment thermal design performance and then to prepare a military specification detailing thermal design evaluation procedures. This paper summarizes the program results with emphasis on the final specification.

INTRODUCTION

Over the past few years, the Armed Forces procurement ag.ncies and airframe contractors have all expressed concern over the increasing cooling provision requirements for aircraft electronic equipment and the aircraft performance penalties resulting therefrom. There has been a general feeling that more could be done to provide optimum thermal design in most equipments. Considerable effort has been devoted to improving the state of the art in electronic cooling; however, surprisingly little has been done in developing specification requirements for evaluating the success of improved thermal design. One of the major reasons for this situation has been the difficulty in arriving at acceptable common denominators of thermal design performance to use as specification parameters.

The program to be discussed herein was set up by Code RAAV-331 of the Bureau of Naval Weapons and the Mechanical Engineering Laboratory of Motorola's Western Military Center with a two-fold objective: first, to define useful and valid measures of electronic equipment thermal performance; and second, to prepare a military specification which will document these thermal evaluation procedures. This paper will summarize this program which culminated in the issuance of MIL-T-23103 (WEP).

PRESENT STATUS OF THERMAL EVALUATION OF ELECTRONIC EQUIPMENT

Definition of the Problem

The subject under consideration is the thermal evaluation of airborne electronic equipment. Since there are no universally accepted ground rules and technology in this field, the first step in this report will be to establish a detailed problem definition and some basic terms for use throughout the subsequent discussion.

It is assumed, initially, that all airborne electronic equipment is designed to perform some specified electrical function in a specified range of environments. The term environment in this case includes everything from dynamic loading to temperature-altitude conditions. It is necessary to incorporate certain provisions into the equipment design to allow satisfactory electrical operation under this variety of expected environmental conditions. As air vehicle performance has increased, the thermal environment in particular has become more severe and there has been a corresponding increase in electrical performance requirements. As a result, the thermal design provisions, or as is typical, the cooling provisions, for airborne electronic equipment, have become more complex with attendant increases in cost in dollars and penalties in flight vehicle performance.

A considerable effort has been devoted to cooling design studies and tests. There has been no general agreement, however, on standard techniques for measuring the actual success or accomplishment of the thermal design effort. The Bureau of Weapons has felt that methods of evaluating the effectiveness of the overall thermal design provisions should be developed and their use required in a military specification.

Study of Thermal Evaluation Provisions in Existing Specifications

The typical current detailed equipment specifications, with reference to the adequacy of their coverage, take in a wide range of thermal problems. Probably the most common situation is the case where only high and low thermal design extremes are specified. These may be quite detailed conditions giving ambient pressure information, coolant flow, temperatures, and so on. On the other hand, many specifications contain only a broad requirement that the equipment operate in some vaguely defined set of "ambient" extremes. The quality assurance or thermal test conditions in the detailed equipment specifications also run the gamut of adequacy. Typically, only a few basic extreme tests are required, with the exact test procedures left open to question. Usually, very little data are required, other than that necessary to establish satisfactory electrical performance of the unit.

Many of the equipment specifications base a part or even their entire thermal environment considerations on the basic general military environmental specifications which include, primarily, MIL-E-5400, MIL-T-5422, MIL-E-5272, and MIL-E-19600A. Here again vague environment classifications based on allowable "ambient" temperatures and pressures are set up with little information and no really accurate definition of the thermal environment. The general test specifications contain a series of combined temperature-altitude test steps. There are no definitive instrumentation instructions nor are the thermal conditions of the test steps completely defined. No provisions are made for external coolant considerations. Very little test data is required, although, a lengthy series of test steps is provided for. MIL-E-19600A contains detailed thermal test provisions, but only on a module basis.

There are exceptions to these general comments, however, there is certainly no uniformity in the presentation of thermal design information or in the required thermal test procedures. In general, most specifications do not provide for a satisfactory evaluation of the thermal design. These specifications are not basically meant to accomplish this. They are concerned with electrical performance and the only point of interest in the thermal environment is to be sure that the equipment will work.

If it is assumed that thermal design is important, then, in view of the foregoing situation, a thermal performance evaluation specification is needed to plug the obvious gap. It must provide organized methods for acquiring, presenting, and interpreting equipment thermal design data. The presentation and interpretation should indicate (1) what the thermal environment limits of operation are, and (2) some measure of relative effectiveness of the thermal design. The information should allow review of the thermal design effort just as the detailed electrical performance data presently required allows evaluation of the equipment electrical design.

There is an important point of differentiation between the specification proposed herein and most of the present general military and detailed equipment specifications. The thermal tests discussed in the latter documents are designed as electrical performance quality assurance provisions. The thermal performance evaluation specification has as its objective the measurements of the relative success of the equipment thermal design effort. Inherent in this is the assumption that the equipment will operate electrically. Therefore, the question is: "What are the thermal operation limits and what did it cost in air vehicle performance penalty and equipment design compromises to provide them?"

THERMAL PERFORMANCE EVALUATION

Any evaluation of a thermal design must first examine the basic parameters incorporated in that design. These are the anticipated thermal environments, the actual physical configuration of the equipment, the required electrical performance in terms of reliability and, finally, the cooling provisions.

TERMINOLOGY

The thermal environment is defined by the specified surrounding air and enclosure temperatures, the ambient pressure and the coolant inlet temperature and flow rate if applicable. Any piece of equipment will operate satisfactorily within some envelope of the foregoing thermal environment conditions. Establishing this envelope is then one primary task of any

thermal performance evaluation study. Typically, one specific set of conditions is considered to be the severest of the likely operational environments. For purposes herein, we will call this specific condition, "The Thermal Design Condition."

The temperature limitations of an electronic equipment are composites of the temperature limits of the individual parts. Each resistor, capacitor, and transistor has associated with it various temperature-electrical operation limitations. There are temperature limits for catastrophic failure, for maximum life, and for acceptable drift to cite a few examples. The basic thermal design problem is to consider these individual part temperature limitations in relation to the specified thermal environments and to provide cooling provisions as required to maintain satisfactory equipment performance and reliability. On this basis, then the thermal environment operational envelope mentioned previously is determined by the maximum allowable temperature of those parts with the lowest temperature ratings. These parts will be subsequently referred to as "critical parts" and are specifically defined as those parts whose surface temperatures are most likely to approach their maximum allowable temperature during the anticipated operating environments.

At this point, it is desirable to digress slightly in order firmly to establish a basis for measuring and discussing part temperatures. No standard means of specifying part thermal environments exists so that one rating may be on the basis of ambient temperature and the next on the basis of maximum internal rise. Basically any part thermal failure or characteristic degradation results from the actual physical temperature of the part. Since all temperatures within a part maintain a more or less fixed relationship with one another, the part surface hot spot temperature becomes a convenient point of measurement and discussion. It is readily accessible and provides reproducible results. The term "part temperature" as used herein will be understood to be part surface hot spot temperature.

The term cooling or thermal design provisions, as used in this report, refers to the fins, baffles, blowers, cold plates, heat exchangers, and tube shields, which provide for some measure of improvement of the thermal environment of the individual parts. For purposes of this study, all electronic equipment cooling provisions can be placed in two general classes:

1. Ambient Cooling. The local external ambient environment is utilized as a major means of dissipating heat from the unit. No external coolant supply is used. Examples are units cooled by external free convection and radiation, and units with external blowers.

2. External Source Cooling. The major heat transfer from the equipment is via some externally supplied coolant source. Examples would include ram air cooled equipments and refrigerated air cooled equipments.

These two classifications make no distinction for specific internal cooling provisions. A sealed, liquid-filled unit with no external cooling provisions and an unsealed unit with forced internal air circulation only, are both in the first class. The major difference between the two classes is the difference in definition of the available medium for heat dissipation.

Basic Approach to Thermal Evaluation

Thermal design, then, is the actual process of considering the specified environment, the part rating data, and the reliability information, and determining the optimum cooling provision configuration. Any evaluation of the thermal design should determine: first, whether the design is adequate to meet the requirements and what its limitations are; and second, whether it is a good design, on the basis of incurred penalty to the air vehicle and overall equipment design. The first step should establish the operating limits for the equipment and this information we shall call "thermal performance limit data." The second step will be referred to as "thermal performance evaluation," and will provide a relative index of the cost of the thermal design to the equipment and air vehicle. Taken together, these steps measure the relative success of the thermal design effort.

THERMAL PERFORMANCE LIMIT DATA

These data will provide complete information on the thermal operating limits and cooling provision requirements for the equipment. The basis for these maximum operating limits is logically the previously discussed part temperature limitations, specifically, the critical part temperature limitations. Assuming that the cooling provision design was based on some reliability goal, then there are established maximum allowable temperatures for all parts. If the most critical part is determined and the equipment operated to determine the limiting

environment conditions where this part temperature can be maintained, but not exceeded, the results are valid thermal operating limit data. First, the critical parts must be identified, their maximum allowable temperatures specified, and a basis for selection of these temperatures given. The equipment can then be operated at the limiting conditions and thermal performance limit plots prepared from the data.

Ambient Cooling

For a unit with ambient cooling, a data presentation as illustrated in Fig. 1 provides a graphic summary of the unit limiting conditions. The solid line is the locus of the maximum ambient temperature and the pressure altitude conditions which will give the maximum allowable critical part temperature. It defines the acceptable steady-state operating conditions of the unit as those lying within the crosshatched area. The location of the thermal design condition should also be noted.

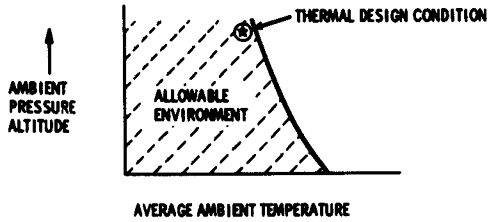

Fig. 1 - Ambient cooling performance limit

External Source Cooling

The steady-state data presentation for units with external source cooling must include coolant flow rate considerations in addition to the external surrounding conditions. In general, it is desirable to know the coolant requirements at various ambient temperatures and pressures. A plot such as Fig. 2 provides the required thermal limit information for a unit with external source cooling under steady-state conditions. This type of presentation is already used in MIL-E-19600A for air-cooled modules; it should have a more general application.

The plot is a limiting condition plot which indicates the coolant flow required at any particular coolant inlet temperature, ambient pressure, and ambient temperature to maintain the most critical part at its allowable maximum temperature. The plot can be produced directly by adjusting the coolant flow to provide the desired critical part temperature at various combinations of ambient conditions and coolant inlet temperatures.

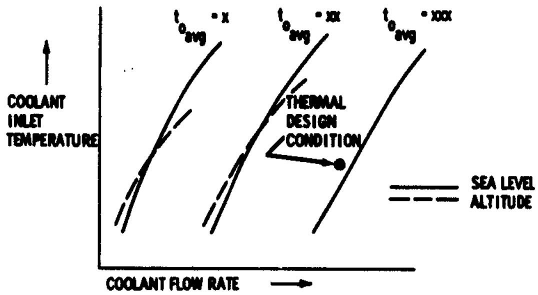

Fig. 2 - External source cooling performance limits

For units using external source cooling, a pressure-drop plot similar to the one in Fig. 3, is necessary. The curve in Fig. 3 was plotted on log-log graph paper to show the corrected unit pressure drop against the coolant weight flow.

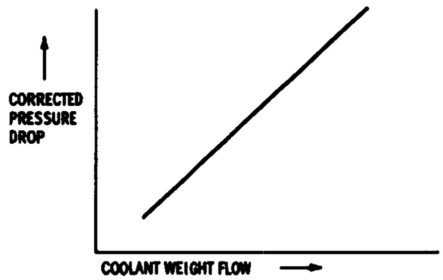

Fig. 3 - Unit flow resistance characteristic curve

Transient Data

Some form of thermal transient information may be of interest for both ambient cooled units and units with external source cooling. Since there are a wide variety of possible situations, only a general plot similar to Fig. 4 can be suggested. This will provide an indication of the temperature rise of various critical components versus time.

In many cases, the actual anticipated operating environments differ considerably from the required test conditions. The present

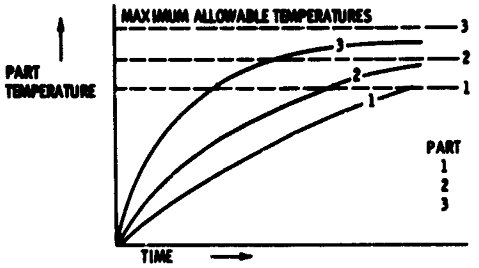

Fig. 4 - Unit transient performance curve

required testing approaches provide insufficient data to make any adequate performance predictions. However, the thermal test data, obtained and presented as discussed, will provide for convenient extrapolation to a wide variety of actual operating environments.

THERMAL PERFORMANCE EVALUATION

If the basic thermal performance limits are established, the next question that arises is, "How good is the design?" There are, of course, a number of qualitative statements that can always be made, but the desired end in this program is a valid formal design evaluation technique. A basic goal of any thermal design provision is to provide the desired environment for the equipment parts with minimum resultant penalty to the air vehicle. Obviously then, the most meaningful evaluation method is one based on aircraft penalty considerations. If it can be shown that the equipment meets the specification thermal requirements by the thermal performance limit data, then the design with minimum resultant penalty to the air vehicle is the most desirable. The following section will deal in detail with the problem of air vehicle penalty considerations.

Aircraft Penalty Considerations

The necessity to provide coolant circulation, electrical power for equipment cooling provisions, and extra structural and fuel weight to carry the additional cooling provision weight, adversely affects air vehicle performance. The actual penalty to the vehicle resulting from an equipment's cooling provisions can be calculated in terms of a basic performance parameter such as miles of range or decrease in payload weight.

Such a penalty parameter should be related to some frame of reference. A comparison of the penalty associated with only the cooling provisions to the total penalty for the entire equipment, provides such a reference. The ratio of the two parameters is essentially a ratio of the "cost" of the cooling provisions to the total "cost" in vehicle performance, of the entire equipment. In order to obtain this ratio, a method of estimating the relative penalties associated with the cooling provisions and the equipment as a whole must be formulated. The first step is to identify those cooling provision factors and overall equipment factors which may affect the air vehicle performance.

Cooling Provision Weight (W_c) — The weight of the items associated with cooling act directly to decrease the payload or the range for the particular vehicle involved. These items include blowers, heat exchangers, baffles and distribution ducts. Even in cases of dual function, i.e., an item such as a cold plate which also acts as a structural member, some portion of the item total weight is chargeable to the cooling provisions.

Cooling Provision Electrical Power Requirements (P_c) — The electrical power required for pumps, blowers, and thermoelectric cooling devices acts to penalize the aircraft by requiring a larger electrical system with attendant decrease in payload or range. This power requirement is another "cost" of cooling to the aircraft.

Cooling Provision Volume (V_c) — The volume required for the cooling provisions absorbs space which could be used for added functional equipment.

Cooling Provision Demand on Central Cooling System (w) — The parameter involves the amount of coolant required at the thermal design condition. In all probability the largest air vehicle performance penalty associated with cooling results from the need for an air vehicle central cooling system. Even the simplest ram air system compromises the air vehicle performance to some degree.

Overall Equipment Factors — The total equipment weight (W_t), volume (V_t), total electrical power dissipation (P_e), and coolant flow (w) affect the air vehicle performance in the same manner as the corresponding specific cooling provision factors discussed above.

Derivation of Simplified Penalty Relations

A preliminary study was made to investigate various methods of thermal design penalty

evaluation. A number of extremely detailed rigorous approaches were tried, however, their complexity did not appear worthwhile in this application. In an effort to define a practical, less rigorous method of penalty evaluation, the following line of reasoning was pursued:

First, it was assumed that we are always considering a flight vehicle whose design parameters are fixed. Even though the vehicle may be in the early design phases at the instant of the equipment penalty evaluation, it is, for that instant, considered a fixed design. This means that no minor changes in any equipment cooling provision parameter will affect the overall performance requirements in terms of mission-range, payload, and so on.

The next assumption was that the flight vehicle was specifically designed to carry a given quantity of payload and equipment on a specified mission. In order to do this, it must provide electrical power, possibly some external coolant supply, and "transportation" for the equipment and payload. We could then define the flight vehicle as essentially being made up of elements to accomplish these tasks.

Based on the concept that these services are equally important to the success of a mission, the weight penalty due to the fuel on board at take-off shall be equally distributed among all the items which make up the complete fuel-less vehicle. It should then be possible to allocate a portion of each of the flight vehicle elements to each item of on-board equipment and payload. In order to accomplish such an apportionment, it is necessary to find a common denominator to represent the flight vehicle items. By assuming that we have a fixed-flight vehicle design with fixed gross weight and range, a logical parameter to use is take-off weight. Each basic flight vehicle element has associated with it a weight at take-off. These weights can be allocated on a usage basis to each of the onboard equipment and payload items. If this allocated weight is then added to their respective weights at take-off, a total equivalent weight at take-off is obtained for each piece of transported equipment and payload. If the allocation can be sufficiently detailed, an equivalent weight at take-off can also be calculated for individual equipment cooling provisions. As previously suggested, this cooling provision equivalent weight can be compared to the equivalent weight at take-off for the entire equipment and a relative cooling provision penalty indication obtained.

The implementation of the previous discussion can be illustrated in equation form as follows: Define basic air vehicle items as follows:

- W_{af} - Weight of basic airframe structure and propulsion group (lb)
- W_{cs} - Weight of flight vehicle coolant supply system at take-off (lb)
- W_e - Weight of all on-board equipment excluding coolant and electrical power systems (lb)
- W_P - Total weight of flight vehicle electrical power plant including distribution system (lb)
- W_f - Flight vehicle fuel weight at take-off (lb)
- W_g - Flight vehicle gross weight at take-off (lb)
- W_1 - Dry weight of flight vehicle (lb)
- P_a - Flight vehicle net electrical power capacity available to items which compose W_e (watts)
- w_a - Flight vehicle coolant supply capacity at flight conditions corresponding to thermal design condition (lb/min)

It follows that:

$$W_1 = W_g - W_f ,$$

and

$$W_e = W_1 - W_{cs} - W_P - W_{af} .$$

The following relations will allow allocation of the take-off weight of the various air vehicle items on a usage basis:

$$\frac{W_P}{P_a} \qquad \frac{\text{lb of take-off weight}}{\text{watt}}$$

$$\frac{W_{cs}}{w_a} \qquad \frac{\text{lb of take-off weight}}{\text{lb per min}}$$

$$\frac{W_{af}}{W_e} \qquad \frac{\text{lb of take-off weight}}{\text{lb of on-board equipment and payload}} .$$

The cooling provision and equipment penalty factors discussed previously will be the basis for allocation of the weight ratio terms to the specific equipments.

If a gross-to-empty weight ratio W_g is included, the total equivalent weight at take-off

or equivalent weight penalty (W_{t_e}) associated with a piece of equipment is defined as follows:

$$W_{t_e} = \frac{W_g}{W_1}\left[W_t + P_e\left(\frac{W_P}{P_a}\right) + W_t\left(\frac{W_{af}}{W_e}\right) + w\left(\frac{W_{cs}}{w_a}\right)\right].$$

Similarly, an equivalent weight penalty (W_{c_e}) for just the equipment cooling provisions can be defined as follows:

$$W_{c_e} = \frac{W_g}{W_1}\left[W_c + P_c\left(\frac{W_P}{P_a}\right) + W_c\left(\frac{W_{af}}{W_e}\right) + w\left(\frac{W_{cs}}{w_a}\right)\right].$$

The ratio between the cooling provision and equipment total equivalent weight penalties W_{c_e}/W_{t_e} provides a valid relative indication of of the "cost" of the unit cooling provisions to the air vehicle.

It should be noted that the cooling provision or equipment volumes do not enter into the penalty considerations mentioned previously. This is a result of the assumption that the aircraft design is fixed. For purposes of this consideration, any change in volume of an equipment would merely reflect as more available space, but would not directly affect aircraft performance as defined by its weight at take-off. The changes could not show up as changes in structure weight or in range, since the air vehicle is assumed fixed and the airframe could not be made smaller or the fuel storage made larger.

Accuracy of Penalty Calculations

The use of penalty evaluation calculations provides the only valid basis for accurately assessing the relative cost of the various unit cooling provision parameters. The above equivalent weight penalty calculation techniques are only approximations of the real air vehicle penalty situation. They are however, simple, and straightforward in interpretation. If the fundamental terms of the relationships are not accurate in absolute value, but are in approximately the correct proportions, both the designer and the evaluator can still make use of the data. The designer desires only to have relative values for his "trade-off" decisions. In order to compare units designed to the same environment, the evaluator will be generally limited to a comparison of units in the same vehicle. The relative values of penalties are used for comparison by alleviating the requirement of absolute accuracy. In the event that the air vehicle manufacturer has a detailed penalty evaluation program set up, it is possible that this program can be utilized to provide the necessary comparative penalty indication.

Reliability Considerations

The fundamental purpose of the thermal design effort has been stated as that of providing for the desired part environment. Essentially, this means providing for part temperatures which are always at or below the temperature necessary to assure proper operation and reliability. In the simplest terms, it is assumed that the equipment must maintain some required reliability or mean time between failures at the thermal design condition. If it does this, the thermal design is adequate from that standpoint and can be evaluated on the basis of its cost to the air vehicle. It is necessary then to demonstrate, as a part of any thermal evaluation, that the equipment will meet the specified reliability requirement. In some detailed equipment specifications there is a firm reliability demonstration requirement. If this takes the form of actual life testing at the thermal design conditions, the test results would provide the necessary reliability verification. In many detailed equipment specifications, however, there is neither a firm reliability requirement nor a reliability test specified. It is felt, therefore, that a reliability determination requirement should be a part of any thermal design evaluation specification. This determination should be made on the basis of the thermal design conditions, to establish what the cooling provisions actually accomplished.

Equipment reliability is usually defined as the probability of its survival for a given interval of time. In equation form:

$$R = e^{-\frac{\tau}{MTBF}}$$

where

MTBF - Mean time between failures (hr)

R - Reliability (probability of survival for τ). (dimensionless)

τ - Specified interval of time (hr)

e - Base of natural logarithms (dimensionless)

This relation is applicable during the so-called constant mean-time-to-failure interval. This period occurs after the initial debugging period and prior to the final wear out. It comprises most of the equipment's useful life.

Determining reliability reduces itself to determining MTBF for the equipment under the

conditions of interest. The straightforward method of obtaining an experimental estimate of MTBF is to operate the equipment over a period of time and to count the failures. The total operating time divided by the total number of failures gives an estimate of the MTBF. The problem is that, to obtain a statistically significant result, a considerable number of failures must be noted and this, in turn, involves a considerable amount of test time and/or equipments. This is not justified merely to obtain a thermal performance bench mark.

An alternate, though somewhat less direct, method presents the most likely compromise. This method involves first measuring part temperatures and electrical loading at the thermal design condition. This information is then used in conjunction with published part failure rate data to predict the expected part failure rates for their actual environment. These failure rates are then combined to give an estimate of overall equipment reliability. A large number of objections to this technique of reliability estimation have been expressed. Despite this, it must be emphasized that the Advisory Group on Reliability in Electronic Equipment (AGREE) supports this approach. The procedure is required by some current Military Specifications such as MIL-R-22256 (Aer) dated 20 November 1959, "Reliability Requirement for Design of Electronic Equipment or Systems." The proposed estimation techniques have also gained sufficient general acceptance to be required by all new contracts for electronic equipment awarded by the Naval Bureau of Weapons.

It is proposed, then, to use actual part temperatures obtained in the thermal design condition test with calculated or measured part failure rate data to provide an equipment reliability performance reference as required for thermal evaluation. If applicable reliability test data is made available, it shall be used for the thermal evaluation.

The foregoing method will provide an estimate of the MTBF for the equipment at the thermal design condition. It remains to relate this to some bench mark to obtain an indication of the effectiveness of the thermal design in providing the desired part environment. The obvious bench mark is the detailed equipment specification requirement, if one exists. In other words, the ratio of the measured or estimated MTBF to the required MTBF should be equal to 1, for optimum design. This ratio defined as $MTBF/MTBF_{spec}$ will be referred to as the Reliability Factor.

It is felt that the emphasis on reliability estimation and the requirement to examine individual parts in relation to their thermal environment will tend to improve equipment thermal design.

Thermal Design Index

On the basis that the Equivalent Weight Penalty Ratio and the Reliability Factor represent the most valid evaluation parameters, they are singled out as the two items defining a thermal design index. This index can be calculated for all equipments as an absolute indication of the success of the thermal design effort in minimizing the air vehicle penalty and attaining the desired reliability performance goal. The index, as such, needed only be presented as two factors:

$$\frac{W_{c_e}}{W_{t_e}} = \frac{(\qquad)}{(\qquad)}$$

and

$$\frac{MTBF}{MTBF_{(specified)}} = \frac{(\qquad)}{(\qquad)}\,.$$

Supplementary Evaluation Factors

The aircraft penalty ratio and reliability factors incorporated into the thermal design index provide a valid absolute thermal design evaluation based on vehicle penalty evaluation considerations. It is, however, also desirable to examine the various individual cooling provision factors on an equipment basis. For example, cooling provision weight can be directly compared to the total equipment weight as an indication of the cost of the thermal design to the equipment. The cooling provision volume was not used for aircraft penalty considerations as previously discussed. It is, however, of interest when examining the detailed equipment design. There are also a number of other factors which will be referred to herein as supplementary evaluation factors which are useful in considering the equipment thermal design. In situations where vehicle data is not readily available, these factors can be used in evaluating the effectiveness of the design. The following supplementary evaluation parameters are felt to be of special interest.

Cooling Provision Weight Ratio W_c/W_t — This factor is simply the ratio of the cooling provision weight to the total equipment weight. In general, a ratio less than 0.15 is desirable.

Cooling Provision Volume Ratio V_c/V_t — This factor is the ratio of the cooling provision

volume to the total equipment volume. The cooling provision volume would include, by definition, the volume of all these items calculated as the elements of the cooling provision weight. A ratio of less than 0.15 is considered desirable.

Cooling Provision Electrical Power Ratio P_c/P_e — This factor is the ratio of the cooling provision power to the total equipment heat dissipation. This ratio would be in the neighborhood of 0.15 for a reasonable design.

Heat Transfer Factor for Unit With Ambient Cooling (HTF) — Along with the separate consideration of various cooling provision factors such as weight, volume, and power, it is highly desirable to examine the detailed heat transfer design. Due to the differences in configuration, a different parameter definition is necessary for units with ambient cooling and those with external source cooling.

For the ambient cooling situation, the parameter selected is an overall heat transfer coefficient between the heat dissipating parts and the external sink. This basic equation for heat dissipation from a unit can be written as:

$$P_e = 144 \text{ (HTF) } A_s (t_{p_{avg}} - t_{o_{avg}})$$

where

A_s - Unit total external surface area (sq ft)

P_e - Unit total electrical power dissipation (watts)

HTF - Heat transfer factor (watts/sq in.-C)

$t_{o_{avg}}$ - Average ambient temperature (C)

$t_{p_{avg}}$ - Average part temperature (C)

Temperature $t_{o_{avg}}$ could be an average ambient temperature or a cold plate mounting temperature. In solving the equation for this HTF we have

$$\text{Heat Transfer Factor (HTF)} = \frac{P_e}{144 A_s (t_{p_{avg}} - t_{o_{avg}})}.$$

The magnitude of this parameter is an indication of the effectiveness of the thermal design. The values used for this calculation should be those obtained from the "Thermal Design Condition Test."

Coolant Utilization Factor for Unit With External Source Cooling (CUF) — The paramount heat transfer design factor of interest in units with external source cooling is the coolant utilization efficiency. This parameter relates the coolant temperature rise to the maximum available potential for coolant heat gain. In equation form:

$$\text{Coolant Utilization Factor (CUF)} = \frac{0.0316 P_e}{wc(t_{p_{avg}} - t_{c_i})}.$$

where

c_p - Coolant specific heat (BTU/lb-F)

t_{c_i} - Bulk coolant inlet temperature (C)

w - Coolant flow rate (lb/min)

The 0.0316 is a constant added to allow use of convenient units and still provide a dimensionless CUF. The values used in the calculation should be those resulting from the thermal design condition test. Ideally, the CUF parameter has a value of 1.

It is acknowledged that there may be additional parameters of interest. A study of the given parameters, in relation to the thermal design environment and the equipment design, will enable a semi-quantitative evaluation to be made throughout the design program.

TEST PROGRAM

A test program was set up to verify and prove out the techniques in the specification. Four pieces of current military electronics were evaluated in accordance with the various procedures and the results were used to assist in preparation of portions of the specification. The details of this program are discussed in "Thermal Design Evaluation Study, Final Report," Motorola Report WF 2472-2, prepared under Bureau of Weapons Contract NOas 60-6030-c.

THERMAL PERFORMANCE EVALUATION SPECIFICATION

The foregoing discussion has established the need for a thermal design evaluation specification and has dealt with specific approaches to the thermal evaluation of airborne electronic equipment. The actual preparation of MIL-T-23103 (WEP) involved the organization of the

above concepts into military specification format. The specification contains the detailed procedure and techniques required to obtain and present the desired thermal performance limit and performance evaluation data.

The previous discussion presented the background for the major evaluation concepts in the specification and the specification itself can be read for the details. It is desirable at this point to summarize the basic steps in the specification procedure as it would be applied to a piece of electronic equipment.

- First a detailed study is made of the unit to identify heat dissipating and critical parts. Actual electrical stresses are determined for all parts or classes of parts. The data are entered on standard forms.

- Next the unit is thermally and electrically instrumented in accordance with standardized procedures. All heat dissipating and critical parts are monitored. The unit electrical performance is checked before and after instrumentation. The unit is then placed in a test chamber.

- The performance limit testing is carried out. This initially involves steady-state performance limit testing based on the limiting critical part temperature. Transient and Thermal Design Condition Testing will then be conducted as required.

- The data are reduced in accordance with specified standard procedures.

- The performance limit data are presented in standard format, e.g., Figs. 1, 2, 3, and 4, and in suitable tabular format.

- The thermal performance evaluation parameters are presented in tabular form.

CONCLUSIONS

The foregoing discussion has been concerned primarily with a detailed consideration of thermal evaluation procedures as a background to the development of MIL-T-23103 (WEP). It is equally important, however, to emphasize the value of such procedures to a company on an internal basis whether a specification requirement exists or not. The standard use of this type of detailed thermal evaluation techniques will soon educate people concerned with equipment design and development to consider thermal problems at the inception of the design. The requirement for detailed part application information continually alerts designers to potential problems. The standard test procedures eliminate the usual confusion over test and instrumentation techniques and provides an assurance of valid data. Most important, the particular presentation of the results makes it obvious as to what kind of a job was done and provides information to make intelligent changes to improve the design. Motorola has been applying these techniques to equipment with significant success in upgrading thermal performance.

MIL-T-23103 (WEP) will be issued initially as a Bureau of Weapons specification and is presently being coordinated with the cognizant industry and service agencies. It is highly desirable that those personnel involved in building, testing, and using airborne military equipment be aware of this specification, its intent and details.

It is intended that the specification will be as a guide during the equipment design and development phases with the full scale evaluation testing being done on engineering or preproduction models.

Since the first issue of any specification may require changes, a program has been set up to review suggestions and to make appropriate corrections or alterations. All such comments should be submitted to Code RAAV-331 of the Bureau of Naval Weapons, Washington 25, D. C., from which copies are available.

DISCUSSION

J. Weil (IBM Owego): You've done some valuable work and made it look so simple. Unfortunately, we've found other problems. One of the difficulties we have run into is the meaning of the word ambient. For example, in one case we found the surface temperature of a component tested in the standard tests, the 65-degree oven "ambient," to be exactly that of an identical component left out on the bench in the "ambient" lab. I think that your project should include more work on the meaning of the word ambient — the pressures, the temperatures, the location of the component in the air stream. Another thing I would like to suggest is a chapter on the cooling of electronic equipment by radiation to outer space, an area we

are getting into more and more in space work. It's done entirely by conduction and radiation. There is no convection. There is no other cooling.

Mr. Baum: I would like to answer this first question on the ambient in that this is exactly the sort of confusion we hope to allay in the specification. I cannot define, in detail, every term now, but we have defined what we mean by ambient specifically in terms of numbers of thermocouples, placement of thermocouples, averaging techniques to be used to weigh the convection ambient adequately, the radiation ambient, and the conduction ambient. This is a good point and I think we have made an attempt to do this. The question of radiation is a very valid one. We also are working on a follow-up program with the Bureau of Naval Weapons to extend the coverage of the specification beyond what it was originally intended for — airborne military equipment. But we have tried to cover those points which you bring up.

Mr. Weil: I hope you realize that I don't mean to be critical. This is a very difficult problem. One other problem we have run into is that of encapsulation. Sometimes when you encapsulate you make passive components hotter than they would be in your airstream and the active components cooler. This is another problem area — so is this reliability number.

Mr. Baum: We feel that use of the techniques in the specification, because they are standard as far as instrumentation and approach are concerned, will begin to do something about some of the problems which have been mentioned here.

* * *

THE RELATIONSHIP OF MEASURED VIBRATION DATA TO SPECIFICATION CRITERIA*

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The highlights of a successful vibration-data measurement program — from sensor selection and evaluation to data processing — are reviewed. Time-sampled data from missile flights are analyzed, and various errors attendant to spectral analysis are considered. Unique methods of data presentation are discussed also. Finally, the respective accuracies of the data analyses are compared, and the impact of these data on design specifications is noted.

INTRODUCTION

For years vibration measurements have been made on flight and captive tests of missiles and aircraft. Instrumentation has been improved to cover a much broader range of frequencies. At the same time, data processing methods have been improved from the earlier waveform analysis techniques (Ref. 1) of counting low-frequency peaks, measuring amplitudes from oscillographs, and presenting "fly-speck" frequency spectra, to the more modern methods of spectral analysis with swept and fixed filters and even autocorrelation. The result is that testing techniques and specification criteria have evolved from sinusoidal tests in the frequency range below 55 cps to combined sinusoidal and random excitation at frequencies to 2000 cps and higher.

The errors and difficulties that plagued the original data measurement efforts — loss of calibrations, the presence of 60-cycle noise with its hierarchy of harmonics, resonance of accelerometer mountings, recording system linearity, and frequency response — are still noted in today's data, but fortunately to a lesser degree. More subtle types of errors, however, do result from certain methods of analysis of random data. The effect of these errors is not so well known nor so universally understood, although several authors have investigated the effect on spectrum analysis accuracies of the time duration of the data sample, the bandwidth of the analyzer/filter, the sweep rate of the analyzer/filter, and the time constant of the averaging circuit in the spectrum analyzer.

It is our purpose to highlight a successful data measurement program by describing data processing methods and the effects of data processing on spectral accuracies, and by outlining some specialized techniques previously unpublished. Further, we will show that ultra conservative specification criteria are the result of data which are improperly analyzed or interpreted.

DATA RECORDING AND INSTRUMENTATION

The first step in an environmental data gathering program is evaluation and choice of instrumentation. The instrumentation/recording system used in a successful missile-vibration, data-gathering program is shown schematically in Fig. 1. Several tests were performed on each piece of hardware in the system to insure the success of the vibration data-gathering program.

Accelerometers were checked for sensitivity and frequency response by the use of standard calibration techniques; for natural frequency and damping by the use of transient

*This paper was not presented at the Symposium.

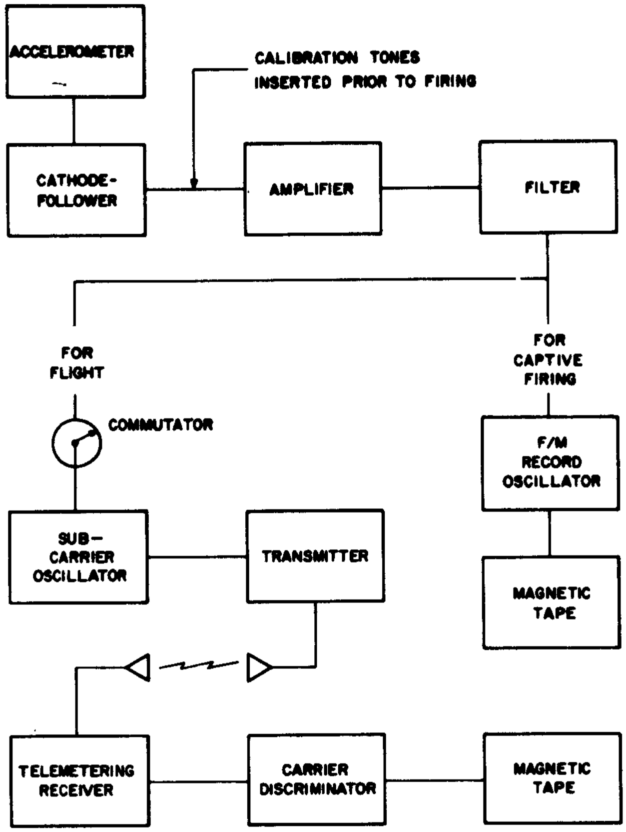

Fig. 1 - Instrumentation recording system

mechanical excitation; for lateral sensitivity or cross-talk by rotating the accelerometers during lateral excitation; and for linearity by the use of resonant beams. Also, calibrations were performed, with and without pads secured to the base of the accelerometer for electrical insulation, to show that there was no change in sensitivity up to 2 kc.

Two noteworthy departures from standard practice were the determination of open circuit sensitivity, S_o, and internal capacitance, C, for each accelerometer and the use of a "dummy" accelerometer.

Open-Circuit Sensitivity S_o, and Internal Capacitance C

These two parameters were determined for each accelerometer. Reference 2 develops the theory and points out the necessity for ascertaining the parameters. If the cable length between accelerometer and cathode follower during service use is different from the cable length during calibration, new accelerometer sensitivities can be calculated using S_o, C, and the capacitance of the replacement cable.

Dummy Accelerometer

On several firings, the output of a cathode follower connected to a dummy accelerometer was recorded. The dummy accelerometer consisted of a 1000-mmf capacitor covered with potting compound shaped to the dimensions and mounting configuration of a standard accelerometer. The capacitor was connected to the cathode follower input with a standard accelerometer/cathode follower connecting cable. This measurement was designed to demonstrate the magnitude and frequency spectrum of spurious noise in the vibration data recording system.

To assure that the dummy accelerometer measurement would not be affected by environmental vibration and would yield satisfactory data on the noise characteristics, the

dummy accelerometer and a cathode follower were vibrated from 10 to 2000 cps with sinusoidal inputs up to ±30 g. There was no noticeable change in quiescent noise.

Cathode followers, amplifiers, and filters each were checked for gain, frequency response, and linearity.

In all cases, sensors were secured to rigid missile structures by the use of solid mounting blocks rather than brackets. Peaks on the data would thus result from an environmental input to the accelerometers which were located as close as possible to the mounting points of electronic equipment. Before several captive firings the accelerometers were excited with transient excitation, and the outputs were recorded to obtain an estimate of natural frequencies. These tests aided in interpreting the spectral characteristics of the measured data.

The filter outputs (Fig. 1) were fed to their respective recording systems and the results recorded on magnetic tape. The data were received in two different magnetic tape formats depending on whether the data were obtained from flight or captive test. In either case, the vibration data were modulated on a standard carrier frequency and once the carrier was removed (demodulated) the method of data processing was roughly the same.

DATA PROCESSING SYSTEM

The data processing system was designed to handle either continuous or slowly sampled (2-second bursts) vibration-data frequency modulated on magnetic tapes.

The following information was needed from the vibration data: (1) variation of rms acceleration, σ, with flight time; (2) variation of zero to peak acceleration (called peak or vector acceleration) with flight time; (3) the probability distribution of instantaneous accelerations in the flight waveforms; (4) the spectral density (g^2 rms/cps) and frequency distribution of the random vibration energy (called the frequency spectrum); (5) the magnitudes (g rms) and frequencies (cps) of periodic components in the waveforms. The data processing scheme was tailored to achieve these objectives.

The data processing system is reasonably standard (Fig. 2), but certain features are noteworthy.

Time Decoder

The time decoder was used to initiate automatically the recording on a tape loop of a preselected time sample of vibration data. It was designed to accept a 10-bit, binary time code, amplitude-modulated on a 1000-cps carrier. A flight time interval was selected for analysis; the time at the beginning of this interval was identified in code; the code was pre-programmed into the decoder; and through coincidence circuitry, the time to start recording was identified. Recording was initiated automatically.

RMS Measurement

To provide checks on the fidelity of recording and the accuracy of data processing, rms accelerations were measured in several different ways:

1. The filtered discriminator output from the original tape was measured on a true root-mean-square (rms) meter for quick-look data.

2. The same output was rectified and fed through true rms detectors whose dc output was recorded continuously on oscillographs to help locate a time interval with reasonably stationary vibration for subsequent spectral analysis.

3. The rms level of the tape-loop discriminator output was measured with a true rms meter to assure that spurious noise had not been introduced in playback and re-recording.

4. The square root of the integral of the frequency spectrum was compared with the measurements 1 to 3 for a check on overall spectral accuracy. With proper care, it was found that these four rms measurements would differ by less than 10 percent.

Choice of Length of Data Sample

When flight data were commutated at a fixed slow-sampling rate, the length of the data sample, and thus the tape loop length, was predetermined. When data were continuous, the length of the data sample was extended as much as was consistent with the stationary character of the waveforms and the allowable tape-loop length on the loop transport.

The long sample was chosen to maximize spectral accuracies while still maintaining reasonable spectral resolution, i.e., using narrow filter bandwidths for definition of resonances.

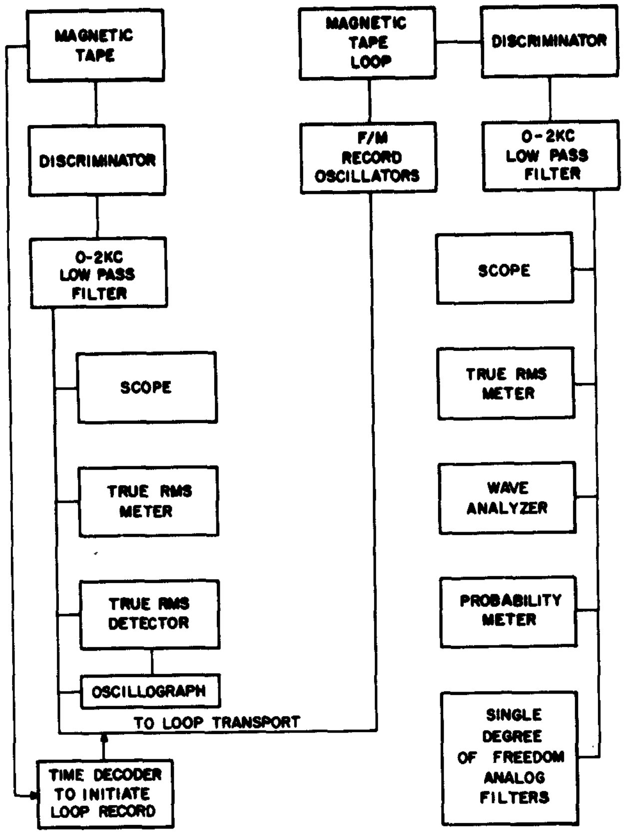

Fig. 2 - Data processing system

References 3, 4, and 5, among others, point out the incompatibility of high spectral accuracy and good spectral resolution. These same references explain the reasons for inaccuracies in power spectrum measurements. Two errors are involved: the error in the quality of the estimate, i.e., the confidence that the result is within the desired approximation of the true power spectrum, and the error in measurement, i.e., the experimental error introduced into the measurement itself. The sample length choice is concerned with the first of these, while the rms checks described earlier help assess the experimental error.

Since the analysis time was finite, the result of each analysis can only be an estimate of the true spectral density. The accuracy of this estimate depends on the product of the sample duration in seconds (T) and the analyzer/filter effective noise bandwidth (B). As the product (BT) increases, the spectral accuracy also increases. For this reason, the tape-loop sample duration was made as long as possible, and never less than 2 seconds.

Choice of Filter Bandwidth

The relationship among filter bandwidth, sample duration, and spectral accuracy is shown in Fig. 3 (Ref. 3). A spectral accuracy criterion was established arbitrarily. It was decided that with 90 percent assurance, errors in spectral density should be less than ±20 percent. Figure 3 shows that the product of sample duration and filter bandwidth should be approximately 70 to satisfy this criterion. Since the shortest sample duration (tape-loop) was 2 seconds, a filter bandwidth of approximately 35 cps was chosen for initial spectrum analyses. Those portions of the resulting spectrum showing relatively high spectral peaks, either from periodic components or resonances of the structure on which sensors were mounted, were re-analyzed with a much narrower filter bandwidth to provide better spectral resolution.

Choice of Filter Sweep Rate

Several authors have investigated the reduction in spectral amplitude caused by

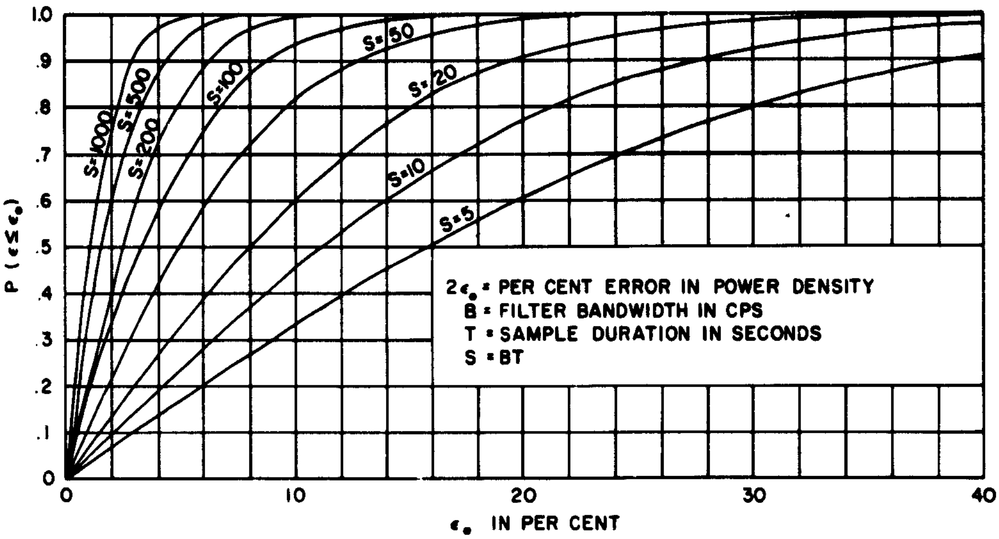

Fig. 3 - Probability of not exceeding error, ϵ_0

filter sweep rate, ϵ. Reference 4 presents the work of one author as:

$$\epsilon_{max} = 2(\Delta F)^2,$$

where

ϵ_{max} is the maximum scanning rate (cps/second), and

ΔF is the filter bandwidth at the half-power points (cps).

The result was determined experimentally on the basis that filter transients would not interfere with the analysis. Chang (Ref. 5) suggests a sweep rate of $\epsilon \ll B^2$ and Moody (Ref. 6) recommends that $\epsilon \leq B^2/4$, where B is the effective noise bandwidth of the filter.

Another criterion for developing an upper limit on sweep rate preserves the statistical accuracy inherent from the data sample length and chosen filter bandwidth. In this case, the sweep rate would be such that the full duration of the data sample is presented to the filter and

$$\epsilon = \frac{B}{T}.$$

This criterion, which is even more conservative, was used for the filter bandwidths available from our analyzer (4 cps minimum) because it presented no hardships in analysis time. In particular, a 2-second data sample could be analyzed to 2000 cps with a 35-cps filter bandwidth in 114 seconds. This sweep rate is slow enough to have no appreciable effect on spectral accuracy for the B and T previously mentioned.

Choice of the Time Constant, τ, in the Smoothing Filter

To minimize the ripple error on the output of the smoothing filter, one would intuitively make τ as large as possible. Reference 7, for example, presents formulas for the ripple error. These are derived from several different types of smoothing filters used on the output of various analyzer bandpass filters. Reference 8 discusses the ripple error of different analyzer filters with outputs smoothed by a simple RC low-pass filter. If the output of a single-tuned analyzer filter is squared and smoothed by a simple RC network, the rms error, e, is given in Refs. 7 and 8 as

$$e = \frac{1}{\sqrt{2\pi B\tau}}.$$

Our analyzer was of this type. Assuming the ripple to be normally distributed, the same confidence criterion established previously was used to determine τ. That is, 90 percent certainty that e is less than 20 percent means that

$$1.64\, e = 0.2 \text{ and } e = 0.122,$$

where 1.64 is the normal deviate corresponding to 90 percent. Using $e = 0.122$ and $B = 35$ cps, gives $\tau = 0.31$ second. The actual components used gave $\tau = 0.47$ second.

Unfortunately, another error, e_s, which tends to limit the value of τ selected is introduced by the sweep rate of the analyzer in scanning spectral peaks. Ratz (Ref. 8) calls this error the bandwidth error in the smoothing circuit and develops a formula from which the following is derived:

$$e_s = 1.386\left(\frac{\tau\epsilon^2}{B_m},\right)$$

where

e_s is the fractional increase in bandwidth,

τ is the smoothing filter time constant (sec),

ϵ is the analyzer sweep rate (cps/sec), and

B_m is the bandwidth of the narrowest spectral peak expected from the random signal under study (cps).

In the section on Choice of Filter Sweep Rate on page 52, the maximum sweep rate was taken as $B/T = 35/2 = 17.5$ cps/sec, $\tau = 0.47$ second, and

$$e_s = \frac{94.5}{B_m^2}.$$

For $B_m > B$ as it should be, the error is less than 7.7 percent. This error can be made negligible by slowing the sweep rate and increasing the data reduction time.

Calibration of the Analyzer

If the spectrum analyzer uses a square-law detector, the dc voltage which causes deflection of the pen recorder is directly proportional to the true mean square voltage out of the analyzer-filter. Thus the pen deflection does not depend on the probability density of the input amplitudes, and either sine or random stimuli may be used to calibrate or set up the analyzer.

Considerably more care must be exercised if the analyzer employs a linear detector for averaging. In this case, a knowledge of the probability density function of the input amplitude is required.

Suppose, for example, that sinusoidal calibrations are used to set up a spectrum analyzer employing linear detection. The scale factor, k, which is the ratio of the rms value of the input to the averager to the recorded dc signal, is

$$k = \frac{\sigma}{v_0} = \frac{A}{\sqrt{2}} \quad \frac{2AG}{\pi} = \frac{\pi}{2\sqrt{2G}} = \frac{1.11}{G},$$

where

G is a gain constant of the averager,

A is the peak voltage of the applied sine wave,

σ is the rms level of the applied waveform, and

v_0 is the dc voltage out of the averager.

For Gaussian random inputs to the linear detector

$$k = \frac{\sigma}{v_0} \quad \frac{1}{G}\sqrt{\frac{\pi}{2}} = \frac{1.252}{G},$$

and the pen read out would be in error by about 13 percent with the random input producing less pen deflection than a sine-wave of the same rms level. (A development of these relationships is presented in the Appendix.)

While the linear detector possesses certain advantages in other applications, only square-law detection was used in our analysis of vibration data, and sinusoidal inputs were used for calibration in setting up the read-out scales of the analyzer.

Separation of Periodic Components

Those portions of the spectra which showed relatively high spectral peaks were analyzed for the possible presence of periodic components by using the results of the wide-band (35 cps) and narrow band (4 cps) spectrum analyses. This method is only approximate since the narrow band analysis is subject to large statistical errors of the type discussed in connection with the length of the sample and the filter bandwidths.

It was assumed that the voltage output, σ_i, in any narrow filter whose width is B and gain is unity, is composed of one periodic component and flat random noise. Then:

$$\sigma_i^2 = \sigma_R^2 + \sigma_P^2 = B_i H + \sigma_P^2,$$

where

σ_i^2 is the total mean square voltage passing through the filter, B_i,

σ_R^2 is the mean square voltage out of the filter resulting from random excitation,

σ_P^2 is the mean square voltage of the periodic excitation,

H is the spectral density of the random noise, and

i is a subscript denoting measurement number 1 or 2.

σ_i was measured for two values of B. From the two equations in the unknowns H and σ_P we find:

$$H = \frac{\sigma_2^2 - \sigma_1^2}{B_2 - B_1},$$

and

$$\sigma_P^2 = \frac{B_2\sigma_1^2 - B_1\sigma_2^2}{B_2 - B_1}.$$

A simple conversion changes these quantities to acceleration spectral density and rms acceleration of the periodic component.

Probability Distribution of Instantaneous Accelerations

A probability meter was designed and built to measure the distribution of instantaneous accelerations in the waveforms. The meter measures, by integration, the length of time that some fixed value of acceleration is exceeded during a fixed time sample of data. The process is repeated as the value of acceleration increases from zero to the largest acceleration in the sample. In this manner, the cumulative distribution of instantaneous accelerations is obtained. The derivative of this curve is the desired probability density function of amplitude.

Analog Filters

It is well known that the performance of a series L, C, R circuit and a mechanical mass-spring-damper system is governed by the same differential equations. This principle was used to examine the behavior of components internal to the package during a flight or captive test.

Circuits which are the electrical analogs of mechanical single degree-of-freedom systems were also built. Provisions were included for varying the quantities that are the electrical analogs of the mechanical parameters of undamped natural frequency, f_n, and transmissibility, Q, at $f = f_n$.

Thus, the f_n and Q of several components mounted within a package would be known from laboratory test data. The f_n and Q associated with these components were selected in the analog system, the appropriate accelerometer data track was played through the analog system, and the mean square voltage proportional to the environmental mean square acceleration response of the component was recorded automatically.

These same filters could be switched from a low-pass to a band-pass operation. When switched in this manner, they provided a check on the validity of the spectrum analysis as determined from the spectrum analyzer. By using several fixed filters to cover the frequency range to 2 kc, any errors from the analyzer sweep rate could be eliminated.

Error Summary

The foregoing discussion indicates that the primary sources of error in spectral density are the statistical error (Fig. 3) and the ripple error (Choice of the Time Constant, τ, in the Smoothing Filter). Thus, for our system and the errors described previously, the probable error is

$$e_{tot} = \sqrt{\frac{1}{BT} + \frac{1}{(2\pi B\tau)}} = 15.5\%.$$

This error can be minimized in an ideal system by:

- Obtaining a very large, stationary, data sample, T;
- Analyzing it with a wide filter bandwidth, B;
- Using fixed, rather than swept filters ($\epsilon = 0$);
- Increasing the time constant of the averaging filters, τ; and
- Making the fixed filters symmetric and rectangular.

CHARACTERISTICS OF MEASURED DATA

The previous two sections indicated some of the factors which affect the validity of measured vibration data. Considerable attention was focused on these factors since each bears on the validity, accuracy, and characteristics of the vibration data which is recorded, measured,

• The magnitude of the noise remained relatively constant with firing time and was independent of changes in the environmental vibration levels.

• The frequency spectrum of the dummy measurement was approximately the same as the frequency spectrum of a data channel before mainstage.

It is interesting to note how such an analysis shows "data" at certain frequencies to be primarily, if not totally, noise.

The data were composed primarily of random noise with periodic or quasi-periodic components superimposed at certain frequencies. These frequencies were generally identifiable in terms of known forcing functions such as turbine rotation, acoustical coincidence (a condition when the wave length of radiated sound coincides with the wave length of flexural vibrations), engine gimbaling, missile free bending, and so on. Thus a test which would demonstrate design adequacy for such an application should logically include shaped random excitation and superimposed sinusoidal excitation.

It is extremely important that the sinusoids be removed from the data and the spectral peaks be correspondingly reduced (Fig. 5, double crosshatch) before generating specification criteria. The fallacy of specifying a random test on the basis of data which includes sinusoidal components is shown in Fig. 6. Level S_1 might be specified, when in reality level S_2 in conjunction with a swept sine is a more realistic requirement. The circles in Fig. 6 at 230 and 580 cps show the reduction in spectral amplitude caused by removal of periodic components.

Figure 6, which was taken from an actual missile flight, shows how short data samples incorrectly analyzed and interpreted can also affect specification criteria. The time sample was only 0.4 second long and was analyzed with a filter of only 6.6-cps bandwidth. The rms statistical error in spectral density is approximately

$$e = \frac{1}{\sqrt{BT}} \times \frac{1}{\sqrt{(6.6)(0.4)}} = \pm 61.5\%$$

and there is still a 32 percent chance that the spectrum presented will differ from the true density by more than 61.5 percent. Increasing the bandwidth to 34 cps reduces the error to about ±27 percent, but much of the spectral definition is lost. The problem lies in the extremely short data sample. Surely, it is not proper to connect the spectral peaks (curve S_1) shown by the narrow-band analysis to generate random vibration specification criteria.

Another interesting presentation of data obtained from a captive firing is shown in Fig. 7. This three-dimensional plot shows that there was very little redistribution in frequency of vibration energy as the firing progressed. As one might expect from examining the change in the causative mechanisms with flight time, the same was not true of free flight.

Distribution of Amplitudes

The distribution of instantaneous accelerations in the flight and captive test waveforms was generally normal (Gaussian) to levels of $\pm 3\sigma$ or three times the rms accelerations (Fig. 8). The only exceptions arose when large

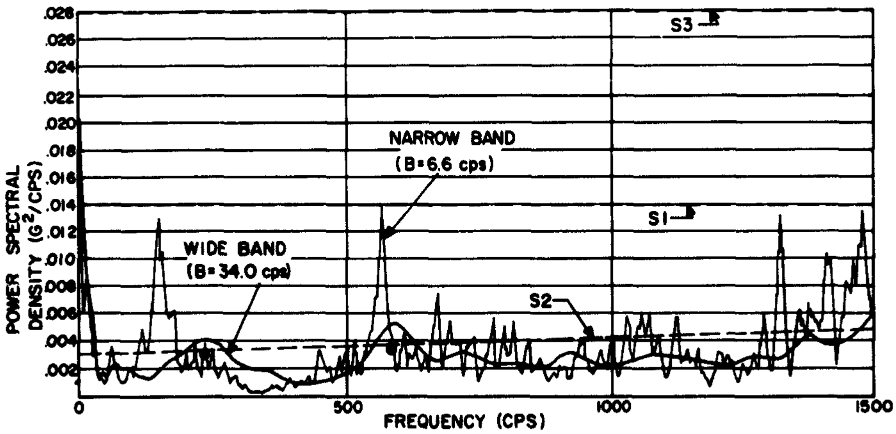

Fig. 6 - Sinusoidal peaks superimposed on random data

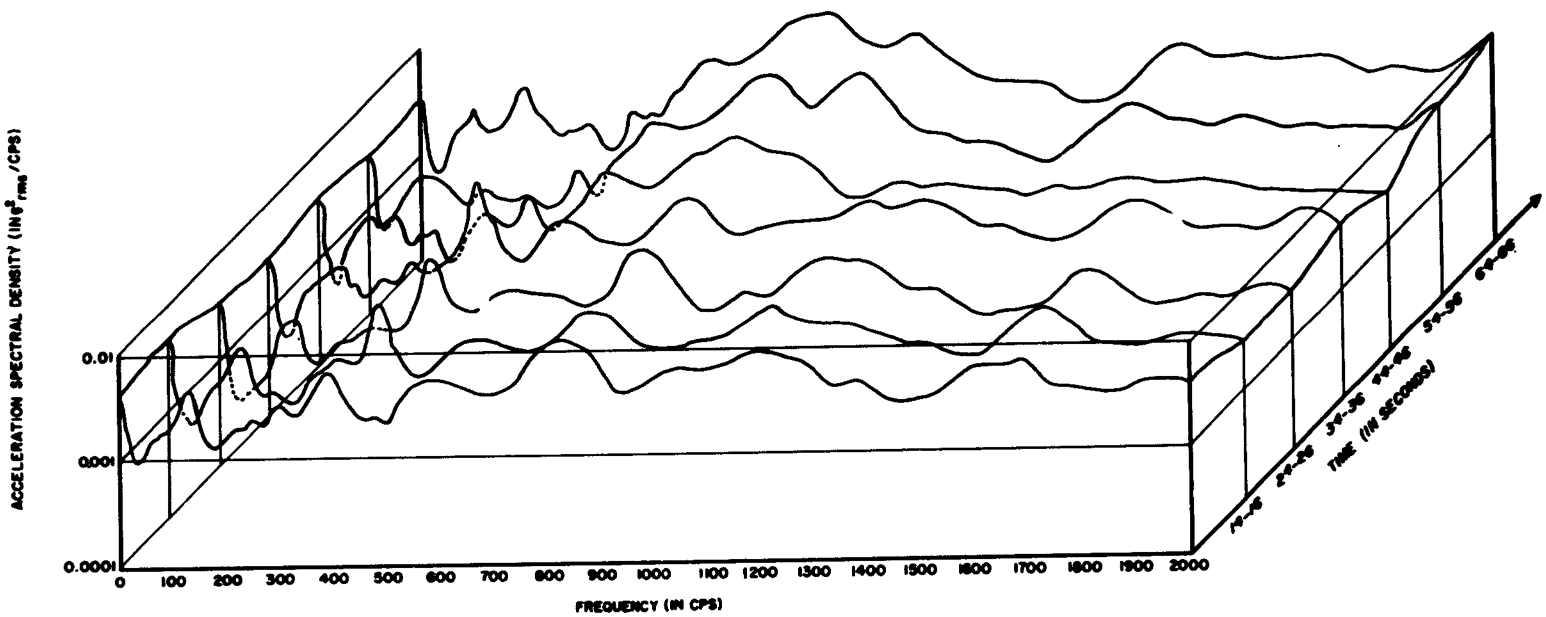

Fig. 7 - Frequency spectra versus firing time

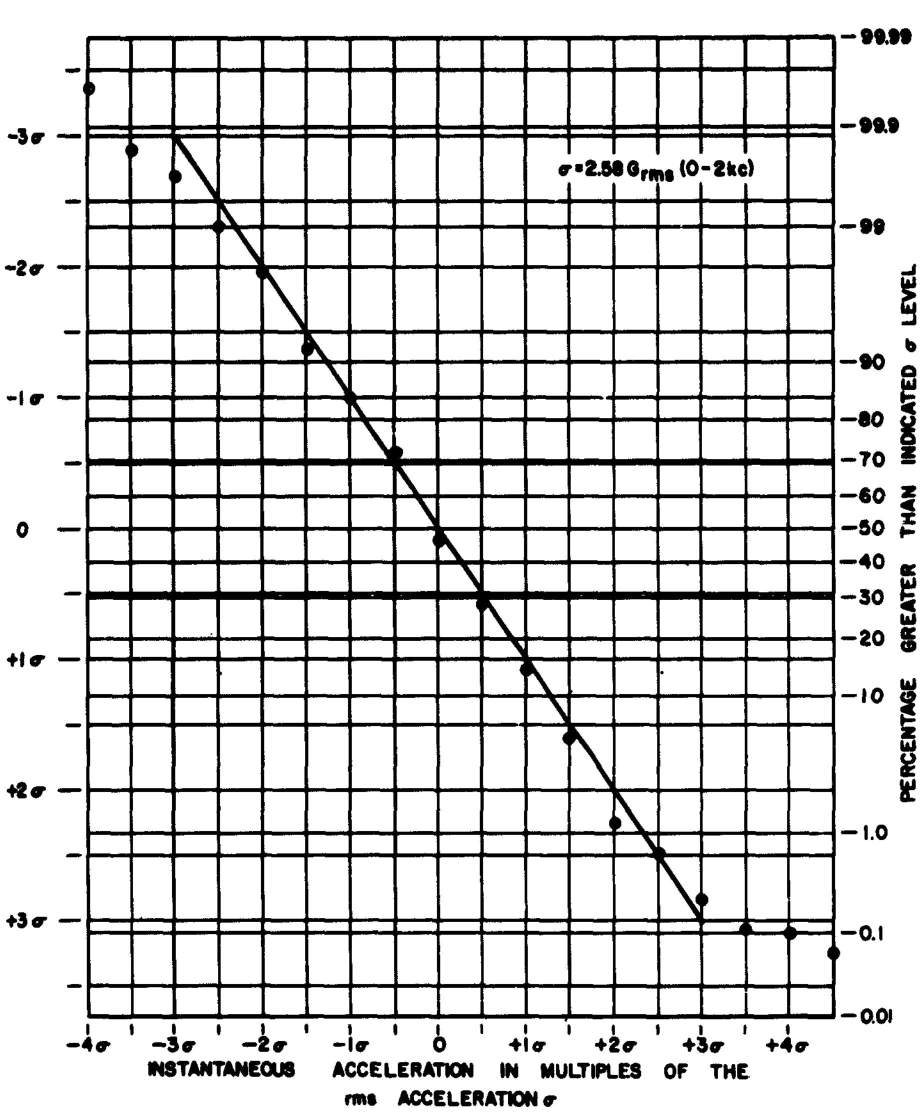

Fig. 8 - Amplitude distribution

sinusoidal components were contained in the data.

At levels greater than ±3 σ, errors were introduced into the probability meter by high amplitude spurious electrical signals which result from dropout and the splice transient on each tape loop. That is, the vibration data is greater than ±3σ for a much shorter time than spurious signals, and errors are introduced which flatten the ends of the curves and tend to give them an "S" shape.

One interpretation of the normality of the distribution is that in a flight of 1000-seconds (16.7 minutes) duration, less than 3 seconds of vibration would contain peaks exceeding three times the rms acceleration. In approximately 50 seconds of vibration the peaks exceed twice the rms level.

Since the output of the random noise generators used in vibration testing and, indeed, the instantaneous accelerations of the shaker table in response to such a stimulus are similarly distributed, the random vibration test is a faithful reproduction of flight conditions in this regard.

GENERATION OF SPECIFICATION CRITERIA

To be useful, the criteria generated must be related to the type and objective of a particular

In Fig. 6, it must be re-emphasized that the area under the frequency spectrum is mean square acceleration or vibratory energy. There is a tendency to connect the spectral peaks as shown by curve S_1, add a generous safety factor arbitrarily or statistically to account for the limited data, and then specify a vibration test requirement corresponding to curve S_3. While an equipment designed to withstand the exposure S_3 would surely survive the measured flight environment, let us compare test levels with data levels. The area under curve S_3 is about 42 g^2 rms from 0 to 1500 cps or about 6.5 g rms, while the area under the measured spectrum is only 6 g^2 rms or 2.4 g rms over the same frequency range. Furthermore, the measured data were obtained from a flight time interval exhibiting the largest vibration during flight, and these levels persisted for only a very few seconds out of the total flight time.

It should also be remembered that curve S_1 was obtained by connecting the peaks from a narrow band analysis with its basic inaccuracies and that periodic components were not removed from the data. Suppose the peak at 570 cps were caused by a periodic component (as it was in fact). The rms level of the periodic component is only 0.27 g rms. Thus, the broadband excitation of level S_2 combined with a swept sine wave of 0.27 g rms would produce spectral peaks higher than any measured in the data whenever the sweeping sine-wave is within the passband of the analyzer. It is now obvious that random excitation at the level S_3 defines an entirely unrealistic vibration spectrum for either design or test purposes. It would seem more logical to apply the safety factors to the overall rms acceleration and account for the spectral peaks and expected frequency shifts of the spectral peaks with swept sine waves or swept narrow-band random excitation.

An often neglected test parameter is the duration of the test. A common approach to the generation of a test specification would call for the application of the Fig. 6 level for the full flight duration. For tests of any length, where fatigue failure is a factor, even small variations in vibration magnitude can have a serious effect on fatigue life. The S-N diagram (stress versus cycles to failure) for any material shows that a small change in stress level will drastically change fatigue life. Fatigue in vibration has been covered in many works. In one of these (Ref. 9), it is shown that a 40-percent increase in acceleration level will reduce fatigue life by a factor of 10. The difference in fatigue life between the 6.5 g rms and 2.4 g rms levels noted in Fig. 6 can be estimated (Ref. 9) to be a factor of 660.

Summary

In summary, certain factors must be evaluated properly before realistic design and test specification criteria are generated from data measurements. Among these are:

- Validity and accuracy of basic data.
- Duration of the various vibration spectra of a flight.
- Establishment of the fundamental objective of the test (for test specifications).
- Establishment of the degree of reliability desired in a design (for design specifications).
- Formulation of realistic safety factors for design and test.
- Penalties involved in the event of inflight malfunction.
- Discrimination of the periodic and random components of the flight spectrum.
- Influence of equipment mounting impedances and reflected impedances from equipment package resonances (dynamic absorption effect).

Consideration of these and related factors will aid the generation of realistic specification criteria. Design and test efforts can then proceed to the development of a reliable product, able to survive its service vibration environment with a pre-determined probability of mission success. Moreover, the weight and performance penalties associated with over-design or the service life uncertainty resulting from over-testing can be avoided.

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Appendix
CALIBRATION OF AN ANALYZER HAVING A LINEAR DETECTOR

If the analyzer/filter output were recorded on an oscillograph, the mean square output, σ^2, could be obtained by squaring the instantaneous voltage output, $v(t)$, and averaging it over time:

$$\sigma^2 = \lim_{T \to \infty} \frac{1}{T} \int_0^T v^2(t)\, dt . \tag{1}$$

If the analyzer/filter output were recorded on a dc recorder, the pen deflection would be proportional to the average (dc) voltage of the waveform, $v(t)$, and not necessarily proportional to σ^2, that is,

$$[v(t)]_{avg} = \bar{v}(t) = [v(t)]_{d.c.}$$

$$= \int_{-\infty}^{+\infty} v\, f(v)\, dv , \tag{2}$$

and $f(v)$ is the expression for the distribution of amplitudes in $v(t)$. The integral definition is an ensemble average of v.

The average value of all symmetric waveforms is zero. Therefore, rectification or squaring and averaging is required if the output is to be read on a dc meter or recorder.

Thus, the problem is to find the relation between the rms and the average or dc value of the waveform $v(t)$ using square or linear detection.

Symbolically, if $\bar{v} = v_{dc} = k\sigma$, k may be different for different forms of $v(t)$. By using the square position and referring to Fig. 9,

$$v_o = G\bar{v}_1 = G\overline{v^2} = G\sigma^2 . \tag{3}$$

In this case, the dc voltage presented to the recorder is directly proportional to the mean square voltage of the analyzer/filter output.

Using the linear position

$$v_o = G\bar{v}_1 = G\,\overline{|v|} \tag{4}$$

where the vertical lines denote absolute value and the bars denote average, but

$$\overline{|v|} = \int_{-\infty}^{+\infty} |v|\, f\left(|v|\right)\, d|v| \tag{5}$$

from Eq. (2).

If we now presume $v(t)$ to be Gaussian, we can write

$$f(v) = \frac{1}{\sqrt{2\pi}\,\sigma}\, e^{-\frac{1}{2}\frac{v^2}{\sigma^2}} \quad \text{for } -\infty \leqq v \leqq +\infty , \tag{6}$$

and

$$v^2 = \sigma^2 .$$

The full-wave rectifier modifies v by taking the absolute value of it and

$$f\left(|v|\right) = \frac{2}{\sqrt{2\pi}\,\sigma}\, e^{-\frac{1}{2}\frac{|v|^2}{\sigma^2}} \quad \text{for } 0 \le v \le +\infty \tag{7}$$

$$f\,|v| = 0 \qquad \text{for } v < 0$$

is the rectifier output amplitude distribution. The factor of 2 appearing in the numerator results from making all negative values of $f(v)$ positive values in $f(|v|)$ and satisfies the condition that the integral of a probability density function over all possible values must be unity.

Now substituting Eq. (7) in Eq. (5) and performing the integration

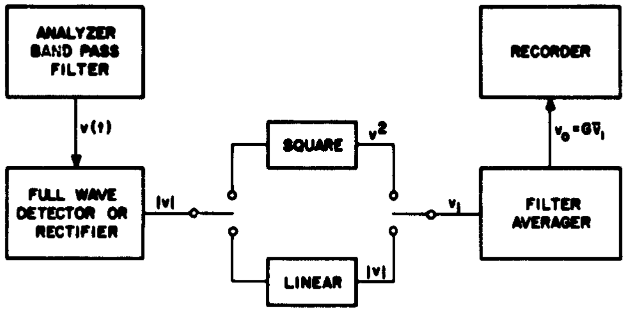

Fig. 9 - Spectrum analyzer schematic

$$\overline{|v|} = \int_0^\infty \frac{2|v|}{\sqrt{2\pi}\,\sigma} e^{-\frac{1}{2}\frac{|v|^2}{\sigma^2}} d|v| \tag{8}$$

$$= \sqrt{\frac{2}{\pi}}\,\sigma .$$

Using Eq. (8) in Eq. (4)

$$v_o = G\sqrt{\frac{2}{\pi}}\,\sigma . \tag{9}$$

With G = unity, the result can be written

$$\sigma^2 = \frac{\pi}{2} v_o^2 , \tag{10}$$

which is the relationship between the rms and the rectified dc output of a random Gaussian waveform averaged with a linear detector.

A similar analysis for sinusoidal v(t) gives

$$v_o = G\,\frac{2\sqrt{2}}{\pi}\,\sigma . \tag{11}$$

Thus, an analyzer with a linear detector calibrated with sinusoidal inputs will read out Gaussian noise inputs incorrectly. For equal rms Gaussian and sinusoidal stimuli, the sinusoidal excitation will produce pen deflections which are 13 percent larger than those caused by the random noise. That is,

$$v_{O.S.} = \frac{2}{\sqrt{\pi}} v_{O.R.} = 1.13\, v_{O.R.} .$$

* * *

PREPARATION AND ANALYSIS OF MUNSON ROAD-TEST TAPES FOR LABORATORY VIBRATION TESTS

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Development and Proof Services
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A program is described to correlate the vibration requirements for vehicular mounted equipment in MIL-E-4970 with the vibration experienced on the Munson test courses. Where the specification levels are shown to be unrealistic, the data from the Munson courses are used to establish realistic vibration tests.

INTRODUCTION

Personnel conducting vibration tests are frequently confronted with a variety of test specifications, the basis for which is often not made clear. The authors became involved in the analysis of the vibration portion of one specification, MIL-E-4970A (USAF) and, as a consequence, tried to study its origin and determine the basis for its parameters.

Manufacturers of air conditioners for the Corps of Engineers had been complaining that their products stood up under Army field conditions, but failed to pass the laboratory tests. In May 1960, the Engineer Research and Development Laboratories (ERDL) of the Corps of Engineers requested Aberdeen Proving Ground to determine what correlation there was between the tests of MIL-E-4970A (USAF) and field vibration tests. If the MIL-tests proved to be unrealistic, Aberdeen was to establish criteria for laboratory tests providing better simulation. The job was to determine the relationship of the laboratory tests to a representative sample of severe field conditions, specifically the Aberdeen Munson Test Courses (Fig. 1). For shock and vibration studies, the following five courses are used most frequently:

- The Belgian Block, Fig. 2(a) (3936 feet)
- The Radial Washboard, Fig. 2(b) (132 feet)
- The Spaced Bump, Fig. 3 (831 feet)
- The 6-inch Washboard, Fig. 4(a) (800 feet)
- The 2-inch Washboard, Fig. 4(b) (821 feet)

During the past 20 years, these courses have been used by other agencies as a basis for developing laboratory specifications for simulated vehicle vibration tests. The original specifications contained in MIL-E-4970 were supposedly derived from tests conducted on these courses in 1948. The Fort Monmouth group of the Signal Corps also obtained data from these courses, which they used to generate their laboratory specification. These specifications were probably too severe. In those days, instrumentation, data gathering techniques, and data analysis were relatively unsophisticated. The job now was to take a "second look" but with the help of most of the latest aids.

Of immediate concern was a family of air conditioners furnished by the Corps of Engineers. They were to be used as representative samples of what the Army might be expected to evaluate prior to acceptance. These air conditioners consisted of one 6000-, one 18,000-, one 38,000-, and one 60,000-Btu/hr unit. The 6000-Btu/hr unit was mounted in a Pershing Communication Pack (Fig. 5), the 18,000- and 38,000-Btu units in an M36 (2.5-ton, 6 x 6 truck), and the 60,000-Btu unit on a 2-wheeled XM419 cargo trailer towed by the truck (Fig. 6). The vehicles used were considered representative of ground transport.

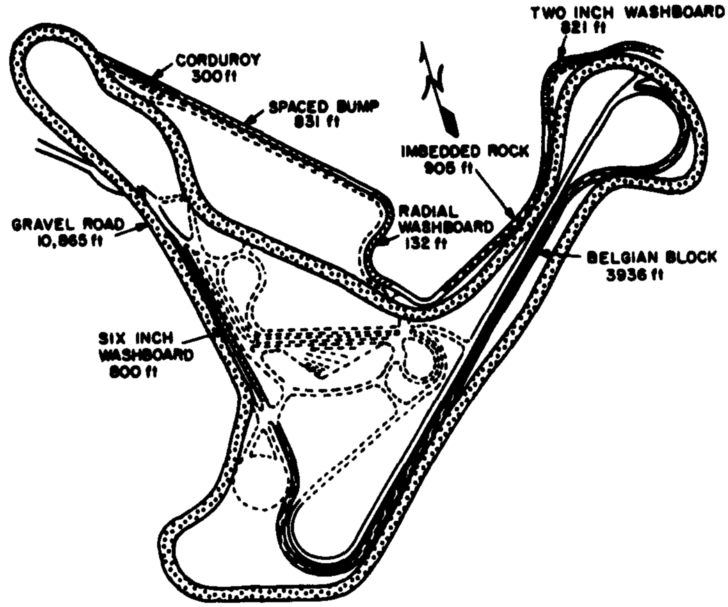
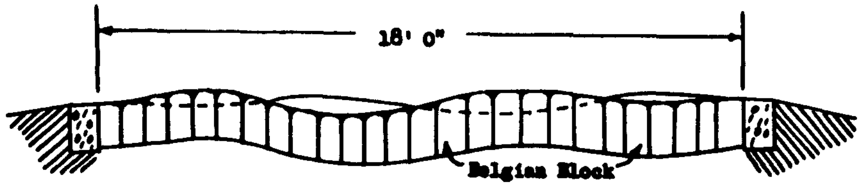
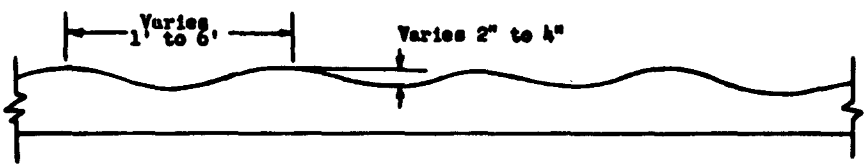


Fig. 1 - Munson test area

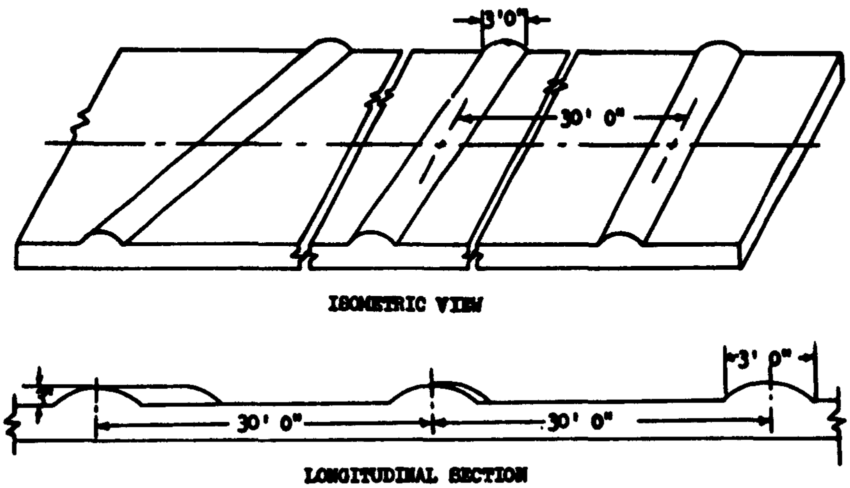


(a) - Belgian Block Course -- This course is a cobblestone road which provides an irregular and bumpy surface. The individual cobblestones average approximately five inches in width. The course irregularities, which not only vary along the length (3936 feet) of the course but also across its width, have crests of about three inches. The crests are such that a vehicle traveling over them is subjected to both pitching and rolling motions.



(b) - Radial Washboard Course -- Two 90 degrees radial turns make up the Radial Washboard Course along with symmetrical bumps which vary from two to four inches in height and from one to six feet from crest to crest. The course is 132 feet long and 20 feet wide.

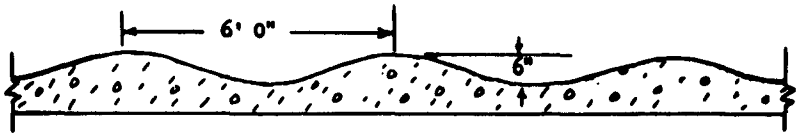
Fig. 2 - Belgian Block and Radial Washboard Courses



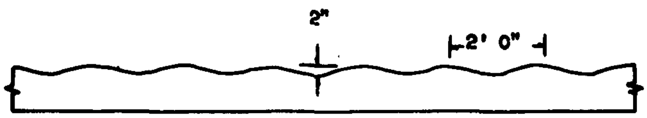
ISOMETRIC VIEW

LONGITUDINAL SECTION

Fig. 3 - Spaced-Bump Course -- This course consists of a series of rounded bumps three inches high by three feet wide spaced at intervals of 30 feet along the centerline of the course. The bumps make the following angles with respect to the centerline of the course: 90°, 90°, 67°, 52°, 90°, 113°, 128°, 90°, 90°, this sequence continues for a total of twenty-six bumps or three cycles for a total of 831 feet.



(a) - Six-Inch Washboard Course -- The profile approaches a sine wave with a double amplitude of six inches and a complete cycle occurring every six feet for a distance of 800 feet. The course surface is concrete.



(b) - Two-Inch Washboard Course -- The profile approaches a sine wave with a double amplitude of two inches and a complete cycle occurring every two feet to a distance of approximately 821 feet. The course surface is concrete.

Fig. 4 - Six-Inch and Two-Inch Washboard Courses

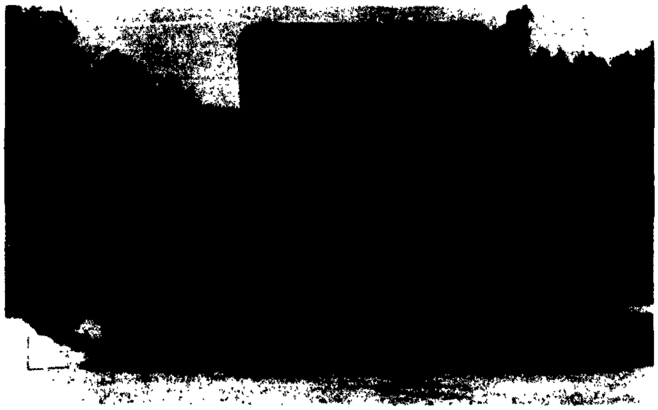


Fig. 5 - Pershing Communication Pack containing the 6000-Btu/hr air conditioner; unit is mounted on an XM474E2 vehicle

INSTRUMENTATION AND TESTS

These vehicle-air conditioner combinations were to be used to obtain representative data with which specification MIL-E-4970A could be evaluated. The object was to place accelerometers on these vehicles in such a way that they would indicate representative inputs to the air conditioners. The response of the air conditioners was not of concern, since they were only samples from any possible supplier. The accelerometers (Statham Model A5A, ±25G) were placed in the locations shown on Figs. 7 through 13. These locations were selected solely on the basis of their input contributions to the air conditioners. No other attempt was made to obtain data on general vehicle motion.

A typical specification for field vibration tests requires that a vehicle be driven at its critical speed over each of the listed courses five times. This critical speed is determined by one of the following: the speed at which the test item is at maximum response; the most severe environment the vehicle driver can tolerate; the speed just below that at which failure of the vehicle suspension or the test item is imminent; or the maximum road speed with respect to vehicle stability. The procedure for traversing the courses is not the same for every test. It may be varied to meet specific requirements. Instead of running five laps with each lap including all five courses, each course may be traversed five times before going on to the next. A transported item generally is inspected visually for damage after each course and, if desired, given an operational check. On two of the five runs, the acceleration inputs and responses are measured and recorded. Prior

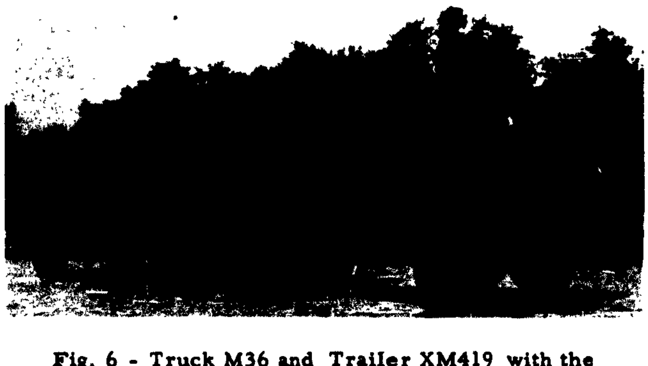


Fig. 6 - Truck M36 and Trailer XM419 with the 18,000-Btu/hr air conditioner inside the shelter, the 38,000-Btu/hr unit mounted externally, and the 60,000-Btu/hr unit

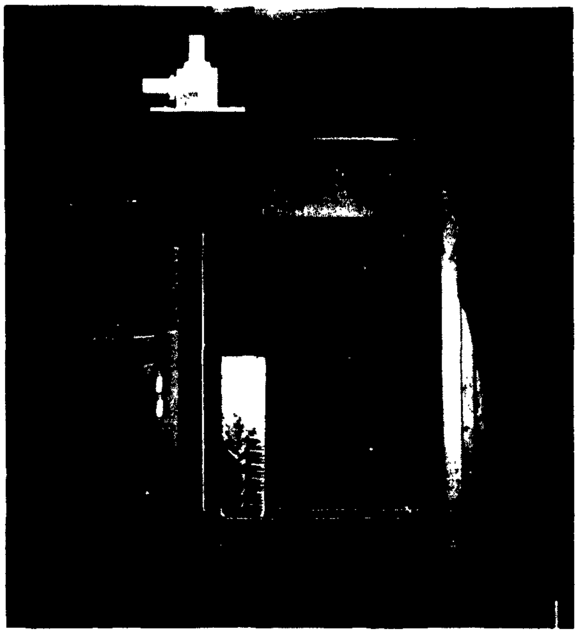


Fig. 7 - Location A: Accelerometers positioned on upper crossmember of 38,000-Btu/hr air conditioner mounting bracket

Fig. 8 - Location B: Accelerometers positioned on the 38,000-Btu/hr air conditioner mounting bracket at left-front of unit (extreme rear of truck)

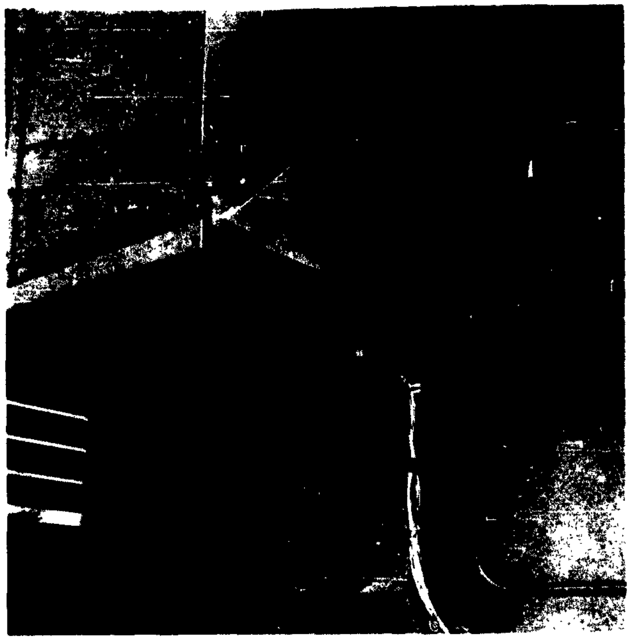


Fig. 9 - Location C: Accelerometers positioned on 38,000-Btu/hr air conditioner mounting bracket at right-rear of unit (adjacent to shelter wall)

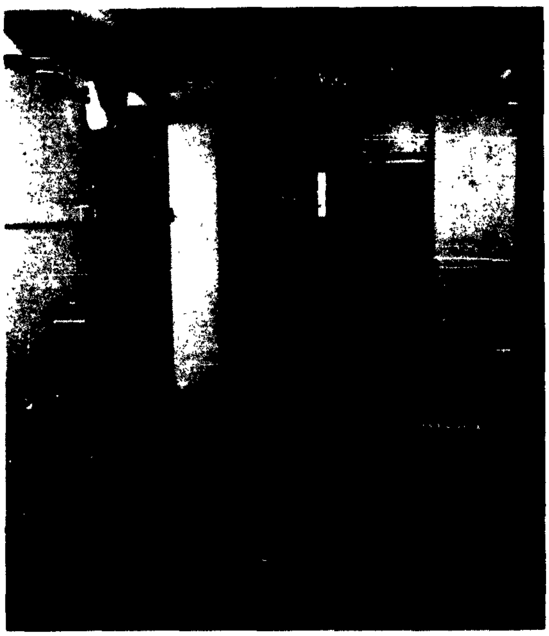


Fig. 10 - Locations E and F: Accelerometers positioned on shelter floor -- foreground location at center of floor, background location at front of shelter adjacent to 18,000-Btu/hr air conditioner

to beginning the first instrumented run, each course is traversed twice to determine critical speeds and recording instrument adjustment. On the course the steps include accelerating to the critical speed, recording data for an interval at the critical speed, and decelerating to leave the course. In addition there is travel over the paved and gravel roads connecting the courses. The three runs not recorded are conducted in the same manner except that the instruments are disconnected and the stops and starts for instrument adjustment are eliminated.

TAPE RECORD

Obviously if data were recorded for the entire operation on the munson test area, a considerable amount of useless data would be accumulated. Interconnecting routes and subcritical speeds would contribute inputs producing stress levels well below those necessary to induce accumulative damage. An attempt was made to produce a tape record which would represent a vehicle traveling at its critical speed over the full length of each course without the intermediate stops, starts, and

Fig. 11 - Location G: Accelerometers positioned on left-rear frame of trailer containing the 60,000-Btu/hr air conditioner

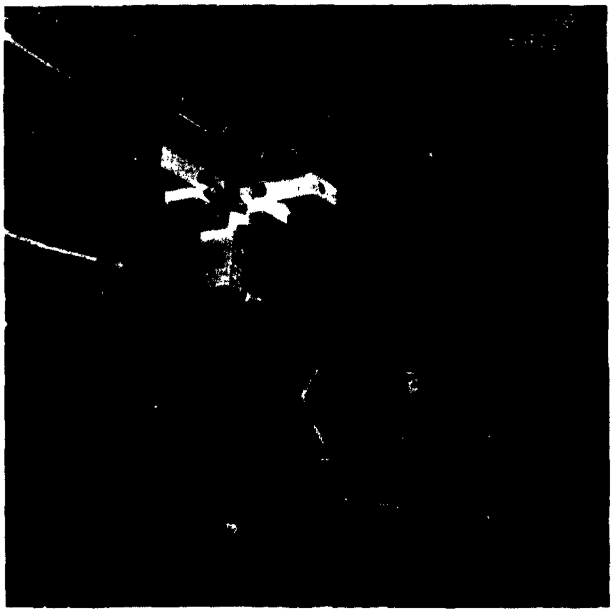


Fig. 12 - Location H: Accelerometers positioned on right-front frame of trailer containing the 60,000-Btu/hr air conditioner

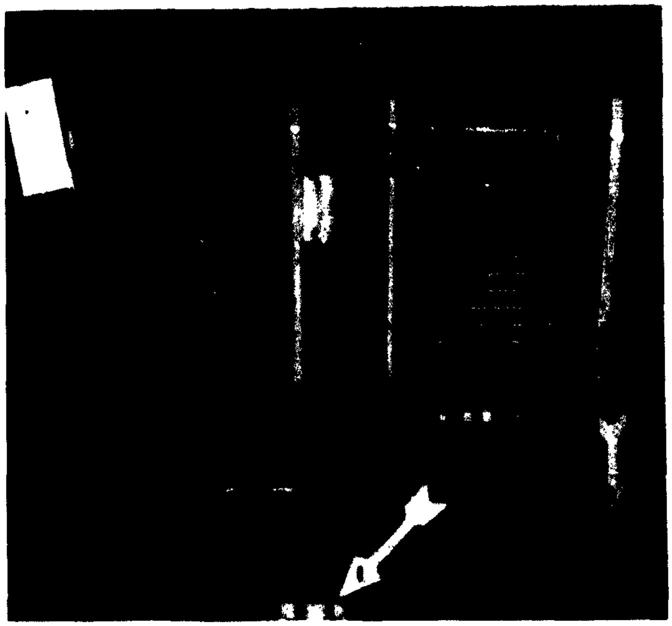


Fig. 13 - Pershing Communications Shelter housing 6000-Btu/hr air conditioners with accelerometers positioned on (1) shelter base adjacent to air conditioners, and (2) bottom of outboard air conditioner above shock isolators

speed changes.* Such a record could be considered a maximum condition for the complete traverse for a given vehicle.

This was a rather complex task since:

- Data samples were taken during only a small portion of the critical speed time on the longer courses.

- A preliminary study had to be made of the records to determine which accelerometers produced maximum g levels, and where in the frequency spectrum they occurred. Signals from these had to be assembled on a tape in such a way that each tape contained a record of the highest g levels reached at any frequency in the recorded spectrum.

- Each course sample had to be replayed on a continuous basis until its re-recorded length represented the full course length.

- The re-recorded data had to be arranged to represent the hypothetical situation of one course following another.

- This tape arrangement in turn had to be reduced from the original by a factor of 16 to be acceptable for data processing.

How these tapes were produced did not necessarily follow sequentially the steps just indicated, for it was possible to perform some of these operations simultaneously. The final tapes for each vehicle were composite records of the maximum inputs to the test items for the vehicle traveling at critical speeds over the Munson test area without the intermediate stops and slow downs between courses. Thus, what is contained on these tapes might be considered end conditions for the Munson test area. The makeup of these tapes is indicated schematically on Figs. 14, 15, and 16. A vehicle traveling under these conditions would move at an average speed of 12.5 mph and would travel over all of these courses in about 6 minutes; the five runs would require 30 minutes.

*Baily, R. D., "First Report on Determination of Shock and Vibration Environment of Air Conditioners Mounted on Trucks, Trailers, or Shelters on Vehicles, Phase I. Development of Tape Loops Representative of the Standard Munson Road Shock and Vibration Test," Physical Test Laboratory, Development and Proof Services, Aberdeen Proving Ground, Maryland, Report No. DPS-450 (January 1962).

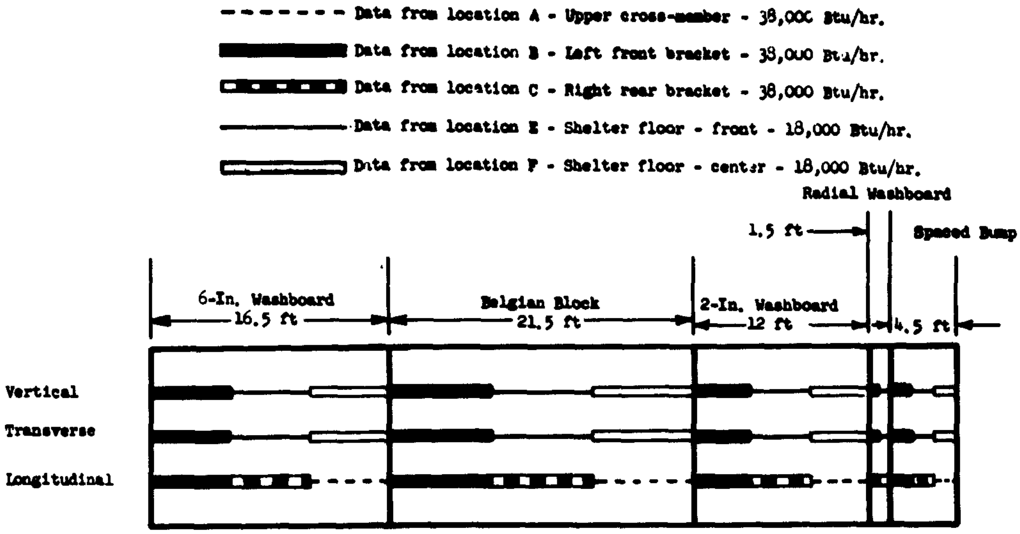


Fig. 14 - Tape Loop No. 1: Accelerometer data from truck-mounted air conditioners -- Munson Course

DISCUSSION

Time-wise, this is a much smaller interval than that of the laboratory tests. Specification MIL-E-4970A, Procedure V requires 12 hours of testing, 4 along each axis. The question arises. On what basis can a comparison be made? For the answer, the general acceptance criteria for army vehicles were studied. Based on Field Force requirements, the Army Ordnance Department specifies that wheeled vehicles should be capable of 20,000 miles of operation without major breakdown with 40 percent of this mileage as cross-country. Track-laying vehicles should be capable of 6000 miles of operation and towed vehicles, 4000 miles. If only cross-country operation is considered as contributory to damage accumulation, there is some criteria for comparison. Accordingly, 8000-mile cross-country was selected as the field condition for correlation. Again, the question arises. What is cross-country? Obviously, this is hard to answer. However, it is hard to imagine more sustained punishment than that

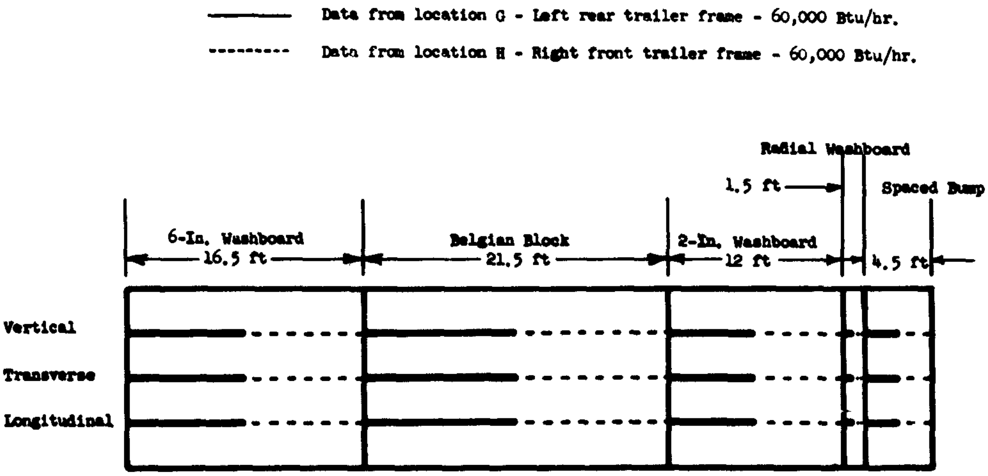


Fig. 15 - Tape Loop No. 2: Accelerometer data from trailer-mounted air conditioner -- Munson Course

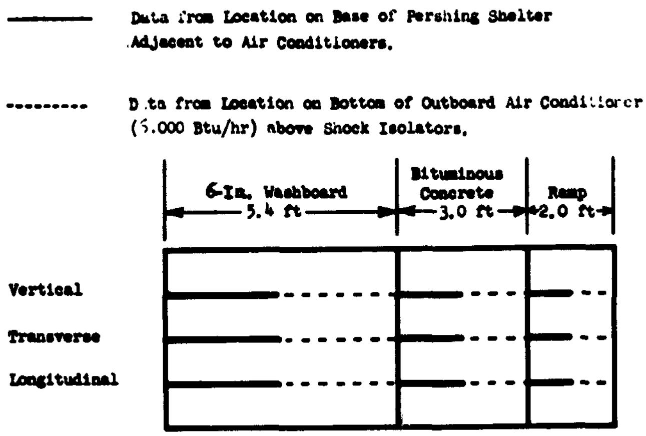


Fig. 16 - Tape Loop No. 3: Accelerometer data from air conditioner mounted in the Pershing Communications Pack -- Perryman Course

experienced by a vehicle traveling 8000 miles under the conditions represented by the composite tape of the Munson test area traverse. Such a vehicle averaging 12.5 mph would travel continuously for 640 hours. The APG film No. ORF-91 on travel over the Munson Test Course is available for circulation. Quite obviously 640 hours is also out of the question for most laboratory testing if random type vibration equipment were available capable of accepting this tape.

If only the events creating damage are considered, however, most of this time is superfluous. One should consider how damage is created. Generally, it occurs as the result of outright breakage, or breakage through fatigue. Some may result from dislocations. At any rate high stress levels are generally associated with damage. Thus, an analysis of the composite tape should indicate the number of occurrences of possible high stress levels; this number may then be projected for 8000 miles of operation. Such an analysis involves generating a spectral density curve for the Munson test area to obtain the g level distribution over the frequency spectrum. From this curve, the location of peak values in the frequency spectrum are found. For most distributions, we can assume, with some justification, that much of the damage occurs at and above the three sigma g levels. Statistical studies have been made of vibration data obtained from these courses by another group at APG. They have determined that the distribution is not truly Gaussian and to establish the number of peaks at and above the three sigma g level, some other distribution curve should be used. This group has determined a number of curves that can be applied.*

Such an analysis of a composite tape is a laborious task, for the records represent a conglomerate of shocks and vibrations. Shock might be produced by a chassis bottoming occasionally on its axles. Shock data should be differentiated from vibration data before measurements of peak values are made. Even measurement of peak values can be misleading. Since each signal may represent the combined effect of a number of frequencies, the signal may be unduly large because of the additive effect of each frequency. Figure 17 curve (1) shows an instance where the peak g recorded was 1.9; curves (2) and (3) show what the g values of the contributing frequencies were at that instant. The maximum value of any frequency in this sample does not exceed 1.1 g. Due to the amount of work involved in this overall analysis, it is not possible to make any more than a cursory comment on the data. That is that the 2.5-g input, specified for both the cycling and resonance portions of MIL-E-4970A, seems higher than any three sigma for any frequency investigated.

It is suspected that, in the past, overall peak values have been used to establish inputs

*Hagan, J., Johnson, R. W., and Tolen, J. A. "Report on a Study Establishing Methodology Describing the Automotive Vehicular Vibration Amplitude Environment," Automotive Engineering Laboratories, Development and Proof Services, Aberdeen Proving Ground, Md., Report No. DPS-657 (August 1962).

for laboratory tests. This may lead to over testing an item, since according to most vibration theory known here, the chief response of a spring mass system is at the frequency of resonance. Thus, if an item has a natural frequency of 50 cps and is subjected, on occasion, to a 5-g input, where this input is the result of the additive effect of more than one frequency, it is unreasonable to assume that the response of the item will be the same as if the 5-g input had been applied at 50 cps. It is intended that tests be conducted to observe the response of spring mass systems to inputs from the combined frequencies found on our tapes.

As previously mentioned, from the standpoint of stress levels, g inputs of low magnitudes have practically no damage potential. Thus, if only those of greatest damage potential are considered, only the levels of occurrence of the peak values should be used to determine the input magnitudes for the laboratory tests. Accordingly, the course that represented the worst Munson area conditions was determined. This happened to be the Spaced Bump Course. This course by composite tape time is traversed in 0.5 minute or about 8.3 percent of the total time. Considering only peak values near and above the 2.5-g level for any discrete frequency, a summation over all the bandwidth frequencies indicated a period of occurrence less than 0.025 percent of the total course time. This percent of the 640 hours required to travel the 8000 miles is approximately 10 minutes. A possible conclusion is that less than 10 minutes of the resonance portion in each direction should be conducted with the 2.5-g input. The rest of the 3-hr period might be more appropriately conducted with a reduced g input. Laboratory tests will be run in the near future to examine these observations.

SUMMARY

The composite tape loops of the standard Munson Road Shock and Vibration Test might be considered representative of maximum conditions for these courses.

The loops can be used for statistical studies and for programming complex wave vibration equipment for laboratory testing.

An analysis of these tapes, using some of the latest techniques and instrumentation, indicates that the 2.5-g input specified in MIL-E-4970A seems higher than any three sigma g level for any frequency investigated.

It is suspected that, in the past, overall peak values have been used to establish input levels for laboratory tests; such procedures would represent over testing since most tests are run at discrete frequencies and these overall values might be caused by the additive effect of many frequencies.

By using 8000-mile cross-country as a basis for comparison of laboratory testing to field testing, a possible conclusion is that less than 10 minutes of the resonance portion in each direction should be conducted at the maximum input. The rest of the period might be conducted at a reduced g level.

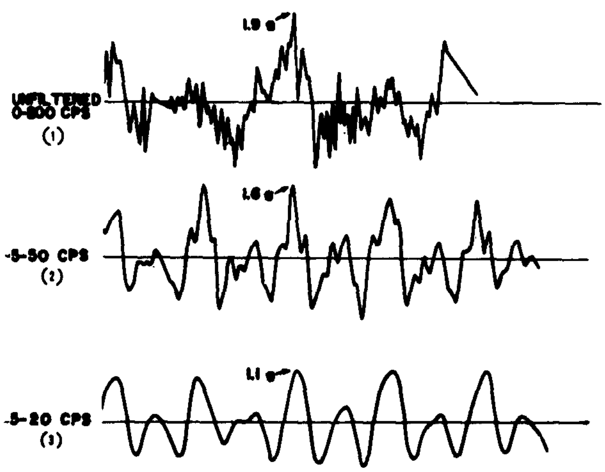


Fig. 17 - Data curve showing additive effect of composite frequencies: (1) unfiltered data 1.9 g; (2) filter passband 5-50 cps, amplitude is 1.6 g; and (3) passband at 5-20 cps, amplitude dropped to 1.1 g

Dr. Mains (GE): I would like to know why you feel it necessary to separate one kind of transient and label it shock, from all the other transients that you label vibration?

Mr. Baily: We were asked to study only the vibration phase of this, maybe later we can get to the shock part of it.

Dr. Mains: How can you tell the difference?

Mr. Baily: At Aberdeen we have a definition for shock that we use in connection with vehicle motion.

Dr. Mains: I think you make a grave error when you do this. You see, if I may try to infer some motives, I would guess that the reason for trying to separate shock from vibration was that if you didn't, then your distributions would be still worse or still farther away from Gaussian than otherwise; I couldn't care less. I haven't yet seen a piece of field data that really fitted Gaussian anyhow, so why try to make it that way?

Mr. Baily: Well, we are not trying to make it Gaussian at all, but we have produced these curves by separating shock from the vibration, and when you introduce shock, as you say, they go all over the place.

Dr. Mains: OK

J. Fowler (STL): I wonder if anyone has proved that the Munson test course is indeed field service or why didn't you take one of your trucks and put it in field service? I'm particularly interested in primary and secondary roads in regards to this, but I am just curious. I sometimes think your course is as false as some of our tests.

Mr. Baily: Do you mean the five courses that we chose? You mean who decided that they were standard tests? There is a gentlemen in the audience who knows a lot more about the Munson courses than I do and that is Mr. Richard Johnson of the Automotive Laboratories. If you have any questions, ask him, I'm sure he would answer them.

Mr. R. Johnson (APG): To answer your question, I think Mr. Baily put it right in your lap. We still don't know what a rough road is — so take your choice. The Munson test courses are in no way intended to duplicate a given virgin territory. We just use them to excite all the resonant frequencies of the vehicle, and nothing else. If your requirements are over secondary roads, you run your tests there. I just want to emphasize that our Proving Ground courses in no way duplicate field conditions.

Mr. Cohen (Sylvania): Did you notice any discrete differences in your recordings between the tracked vehicles and the wheeled vehicles driven over the same course?

Mr. Baily: Quite a bit. The most severe environment for track laying vehicles is smooth roads. You get a more sustained, higher g level. For the wheeled vehicles the most severe environment is the five courses that we used here.

Mr. Cohen: In other words you are saying that the tracked vehicles, over this course, had relatively smoother rides than the wheeled vehicles over the same Washboard or Belgian Block Course, and so on?

Mr. Baily: Yes, I guess you could say that. The frequencies of acceleration are much higher on the track laying vehicles. With these track laying vehicles you get the frequency of the treads and the pads.

W. Studebaker (Collins Radio): I have been in direct association with the communications system which you spoke of and maybe can shed some light on this subject of the difference between wheeled vehicles and tracked vehicles. I think the main difference is not the fact that we receive higher g levels on the tracked vehicle, but the fact that, at the track laying frequency, this g level is sustained for a long period of time. Should this track vehicle travel a course at some given speed, the one in this instance which was most critical was 22 mph which gives us a frequency input of about 64.5 cycles, this input is sustained during the period of time which this vehicle maintains this speed. Should it travel for 1.5 hours at this speed, it means that you have a steady frequency input for 1.5 hours at whatever the input level happens to be, which varies with the hardness of the road course.

Mr. Stern (GE): I noticed that in Fig. 17 you had the different filter settings and then you commented how the acceleration levels dropped. Now, I would imagine if you had, say instead of 5 to 20 you set the filter setting at 5 to 10, it would drop even more, and if you set it from 5 to 8 why the level would be even less. So, what was the significance of that? I didn't get the point to that.

Mr. Baily: Well if you were going to vibrate the thing at 20 cycles you wouldn't use 1.9 g as an input would you? Because 1.9 g didn't occur at that frequency

Mr. Stern: Were you trying to show then that from the 5 to 20 cycle setting was the predominant frequency? You felt this was it or that you could have gone further?

Mr. Baily: We felt that that was it.

Mr. Stern: Well this is arbitrary on your part. You just get down to where it looks nice, you stop there, and then you say that this is about it then.

Mr. Baily: Yes.

Mr. Stern: One final question, since these Washboards all seem to be rather periodic like, say half sine waves, do you ever try to represent the truck and the Washboard on an analog computer before you go to these road tests?

Mr. Baily: This is being considered.

* * *

DEVELOPMENT OF VIBRATION DESIGN PROCEDURES FOR THE ORBITING ASTRONOMICAL OBSERVATORY

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This paper presents analytical and experimental procedures used to assure the adequacy of the spacecraft structure and equipment under the vibration requirements of the NASA Environmental Test Specification.

INTRODUCTION

The Orbiting Astronomical Observatory (OAO) is a precisely stabilized, 3300-pound, scientific satellite capable of accommodating a variety of astronomical experiments. The first observatory will be launched in 1964 by an Atlas-Agena Booster from the Atlantic Missile Range into an approximately circular orbit at an altitude of 500 statute miles.

Each OAO is comprised of two main systems, the spacecraft and the experiment package. The spacecraft is being developed by the Grumman Aircraft Engineering Corporation for the Goddard Space Flight Center of the National Aeronautics and Space Administration. The different scientific experiments, weighing up to 800 pounds, will be supplied to NASA by leading astronomers and observatories, including the Smithsonian Astrophysical Observatory, the University of Wisconsin, Princeton University, and the Goddard Space Flight Center itself.

This paper describes the development of analytical and experimental vibration design procedures used to assure the adequacy of the spacecraft structure and equipment under the requirements of the NASA Test Specification.

DESCRIPTION OF THE OBSERVATORY

The OAO is an aluminum structure 116 inches in height with an octagonal plan form measuring 80 inches across the flats; it encloses a hollow, 48-inch diameter cylinder which houses the experiment package. Solar paddles, folded flat against the octagonal sides during launch, are erected after booster burnout to provide electrical power.

A general arrangement of the OAO structure is given in Fig. 1. The primary structure consists of a central tube, eight trusses emanating from the tube and extending to the apexes of the octagon, and seven shelves or bulkheads connecting the trusses and tube. These shelves divide the spacecraft into 6 toroids, which are further divided by the trusses into 48 bays, wherein all items of equipment, e.g., data processing, stabilization and control, and so on, are located. The spacecraft is attached to the Agena by means of a 14-inch high interstage structure in the form of a conical frustum with a 60-inch diameter at the booster attachment plane.

VIBRATION REQUIREMENTS

The NASA Vibration Test Specification requires that a broadband motion input, at the levels shown in Table 1, be applied to the base of the spacecraft. In recognition of the possibility of generating unrealistically large load factors at spacecraft antiresonant frequencies, the specification states that "during vibration testing, the vibration input be controlled so that, at or near the major modal frequencies, the design strength of the structure shall not be exceeded." The specification further requires that the response of the OAO structure to the base input motion be determined at all equipment locations. This response is then to form the basis of the vibration requirements for the equipment.

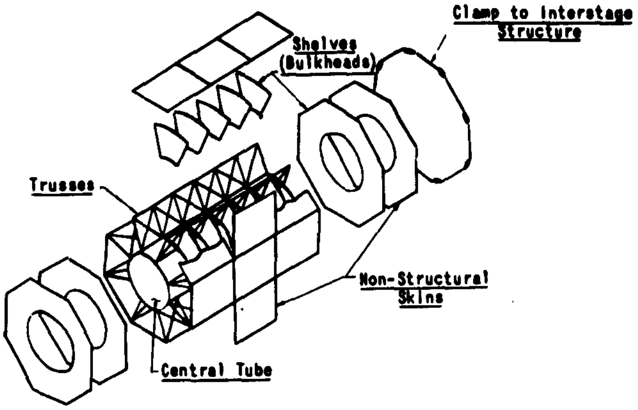


Fig. 1 - Structural arrangement

TABLE 1
OAO Vibration Test Requirements

Sinusoidal		
Axis	Frequency (cps)	Amplitude
Longitudinal	5 - 10 10 - 50[a] 50[a] - 400 400 - 2000	0.5" D.A. 2.5-g Vector 5.0-g Vector 7.5-g Vector
Lateral	5 - 9 9 - 400 400 - 2000	0.5" D. A. 2.0-g Vector 7.5-g Vector
Random		
All	70 - 2000	0.03-g^2/cps, 8 minutes/axis

[a]This frequency shall be adjusted to correspond to 1.414 times the first longitudinal major structural mode of the spacecraft.

ANALYTICAL TREATMENT

Since it was necessary to establish vibration requirements for all items of equipment long before vibration levels could actually be measured during a vibration test of the Observatory, a series of forced-response analyses were performed by considering the OAO a multi-degree-of-freedom cantilever beam system subjected to the motion input.

The structural arrangement of the spacecraft, i.e., seven equipment toroids, suggested a seven mass point representation of the spacecraft structure and equipment mass distribution. In addition, a representation of the experiment package was required. Unfortunately, at the time of the analysis, there was no available information on the mass and stiffness distribution of the experiment. In fact, the experimenters themselves had not yet fixed the configuration or contracted for the construction of the packages. Therefore, the weight of the first experiment was estimated, and the package was considered as a single mass. The lumped-parameter representation of the OAO used in the forced response analysis is shown in Fig. 2.

The influence coefficients needed to represent the structural stiffness distribution were derived from an elastic analysis of the OAO structural arrangement. The analysis was based primarily on the assumption that the equipment shelves were of infinite rigidity, i.e., each shelf was assumed to be completely rigid, moving as a unit. It was further assumed that all modes were uncoupled, i.e., there was no bending with longitudinal or axial motion, nor torsion with lateral or transverse motion.

The analysis was a standard lumped-parameter, normal mode treatment modified to account for a motion input.[1] The equations of motion and the forced response analysis are presented in the Appendix.

It was previously mentioned that the NASA Specification required that input motions be controlled in order to preclude exceeding the design strength of the structure. The shears, bending moments, and axial loads due to the specified inputs were therefore calculated and compared

[1]Unpublished work of William Clark and Louis Pulgrano of the Dynamic Analysis Section, Grumman Aircraft Engineering Corporation.

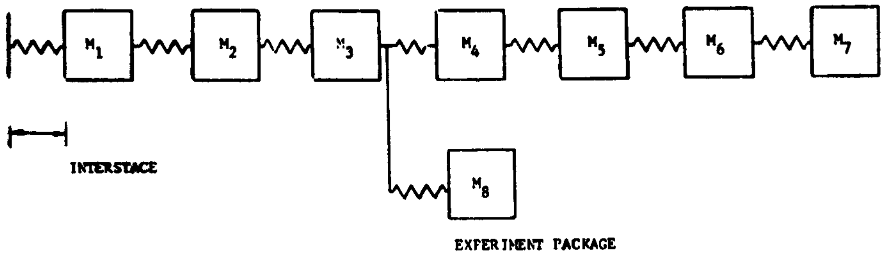


Fig. 2 - Lumped parameter representation of OAO used in analysis

with the static design criteria. At those frequencies where the analysis indicated that the design strength would be exceeded, the maximum allowable input levels were determined. The response of the OAO to this adjusted input spectrum was then calculated, and the motions of equipment and structure at the various locations in the spacecraft were obtained. Figure 3 is a typical example of the results of the analysis. The response is shown only for those frequencies where the basic input to the spacecraft is exceeded. The envelope enclosing the response spectrum is taken as the equipment vibration requirements. This envelope was suggested by the NASA to provide for simplicity in testing, and to allow for the inaccuracies resulting from the simplifying assumptions used in the analysis.

The structural damping coefficient plays a significant role in a vibration analysis such as this, since it is this parameter which limits resonant response. Based on Grumman's experience with airplane structures, a structural damping coefficient (equivalent approximately to twice the critical damping ratio) of 0.04 was chosen as being typical of this type of built-up, riveted aluminum structure. Tests of the structural model later revealed damping factors which were considerably higher than this assumed value. In most cases, however, this assumption did succeed in producing the predicted equipment and structural motions conservatively greater than those which actually occurred.

VIBRATION TESTS

A vibration test program was conducted on the structural model spacecraft in order to establish the structural integrity of the spacecraft. Although, as previously mentioned, the strength of the structure was limited to static design criteria, the vibration tests served to demonstrate the continued alignment of primary structure, the adequacy of secondary structure, equipment supports, and so on, and to provide confidence in both primary and secondary structure under the action of local resonant conditions.

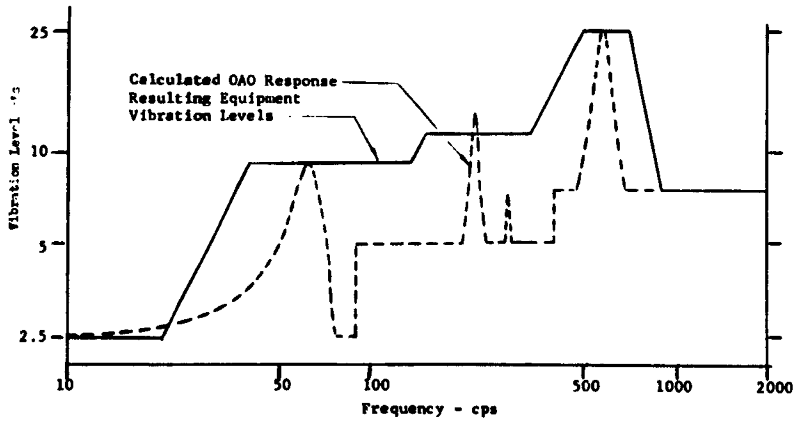


Fig. 3 - Results of vibration analysis

A second objective of the tests was to measure the actual vibratory environment to which equipment and subsystems would be subjected during the vibration tests of the prototype observatory. These measured vibration levels were to supersede those calculated by analysis. They would in addition serve as a check on the analysis.

In order to specify the input levels at frequencies close to the major modes, it was necessary to determine the vibration loads which would just approach the design strength, as required by the NASA specification. These loads were obtained by three independent methods:

- The loads in the spacecraft structure were calculated using the type of normal mode analysis initially used to estimate the environment, except that the actual measured mode shapes and damping coefficients were substituted in place of the previously calculated and assumed values.

- The force applied by the shaker to the base of the spacecraft was determined from armature-coil-current measurements made during the tests, to yield the load (shear or axial) at the base of the spacecraft.

- Strain gages installed on primary structural members and monitored during previous static structural tests were used to limit vibration levels to keep strain values during the vibration tests no greater than those measured during static tests.

During the mode surveys, and later during the actual vibration tests, measurements were made at more than 100 locations in the spacecraft. To keep the loads calculations within reasonable bounds, the mode shapes were "averaged" across each shelf station; this resulted in a 10-mass configuration which was similar to the previous 8-mass system, but with the addition of better experiment package representation.

The structural loads calculations, therefore, also include the original assumptions regarding shelf rigidity and uncoupled modes. The behavior of the OAO observed during the tests was appreciably different. There was considerable elastic motion of the shelves during the longitudinal tests, and a significant amount of torsion in the lateral tests. Figure 4 is an example of "umbrella" motion, as observed during the longitudinal tests, which shows large amounts of shelf deformation. Because this behavior was not considered in the loads analysis, it was felt that the calculated structural loads were unconservative. The strain gages, on the other hand, measured the actual strains in the primary structural members, including the bending and torsional effects. The strain gage responses were therefore chosen as the governing factor in comparison with the loads calculations.

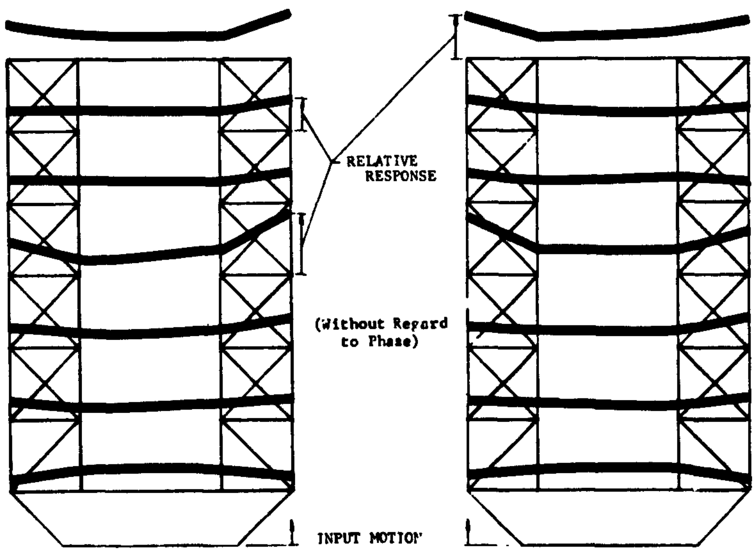


Fig. 4 - OAO longitudinal "umbrella" motion-145 cps; views of orthogonal axes

The use of the shaker-armature-coil-current measurements as a load indicating technique might have been adequate if the OAO response were unidirectional. However, the presence of bending and torsion in the spacecraft primary structure clearly resulted in unknown reactive moments at the shaker support points. This led to the conclusion that the average shaker axial load was not as good an indication of the actual total load in the OAO structure as the strain gages. These measurements, therefore, were also discarded in favor of the strain gage readings. It is interesting to note, however, that all three procedures yielded similar results, differing by only about 30 to 40 percent.

Following the survey of each mode, an attempt was made to measure the modal structural damping coefficient by using the standard frequency bandwidth, or half power point procedure. This method is considered to be valid for distinct, well-separated modes with magnification factors of at least 10. The modes which were measured (the lower three or four in each direction) were, in general, neither distinct nor well separated; they had magnification factors in the range of 5 to 15. Therefore, there was considerable doubt as to the accuracy of this method.

There is, however, a straightforward procedure by which the modal damping coefficient may be calculated by use of known or measured mode shapes, mass distribution, and modal magnification factor. This method is described in the Appendix. Damping coefficients in the range of 0.10 to 0.45 were obtained by this method. These values were experimentally corroborated in those modes on which the frequency bandwidth method could be used.

The extraordinarily high damping values are attributed to the energy dissipation in the many expansion joints incorporated into the equipment support structure. These joints were designed to increase thermal isolation between equipment and spacecraft structure in order to reduce structural misalignment due to thermal stress gradients.

Since the vibration input levels had to be reduced only in those modes where the design strength would be exceeded, only the first three or four modes in each direction were measured. A comparison of these mode shapes with those previously calculated showed good correlation with some exceptions. The interstage proved to be considerably softer in the axial direction than anticipated. Modes were also found in which almost the entire motion was in the experiment package. Due to the original single-mass representation of the experiment package, these modes could not have been quantitatively predicted; however, they were qualitatively predicted, and came as no surprise.

Because of the normal mode type of analysis used in predicting equipment motions during tests, the analysis was invalid beyond the frequency of the highest calculated mode. Within this limitation, reasonably good agreement was found between the measured and predicted modal frequencies. The measured vibration levels, on the other hand, were generally less severe than had been predicted, primarily due to the high structural damping found during the tests. However, high-frequency, moderately-damped motion did occur in the lateral direction where the motion reached 20 to 30 g in the 300- to 500-cps range. This was the only area in which unexpectedly severe vibration levels were encountered.

In addition to the sinusoidal vibration tests described, the NASA specification also required that a random vibration test be conducted on the OAO. The input spectral density is shown in Table 1. The original requirement of including structural amplification in equipment design and test criteria applied to the random as well as the sinusoidal test.

Measurement of the test environment at the many equipment locations necessitated repeating the sinusoidal tests several times in each direction. Application of this procedure to the random vibration tests would have unduly increased the exposure to vibration, thus creating the possibility of unwarranted and unnecessary fatigue failures. Furthermore, it was felt that random vibration requirements for equipment could be determined with reasonable accuracy by applying transfer functions measured during the sinusoidal tests to the specification input levels. Repeated random vibration tests were not conducted; rather, the random tests were performed only once in each direction, for the qualification of structure. During these runs, vibration measurements were taken to check the accuracy of the transfer function procedure. A comparison of the measured transfer function with a spectrum analysis of the acceleration during the random test is shown in Fig. 5. It is apparent that the low-frequency portion of the spectrum is overestimated by using sinusoidal transfer functions.

TEST VERSUS FLIGHT ENVIRONMENT

The purpose of prototype vibration tests is to provide assurance of successful operation

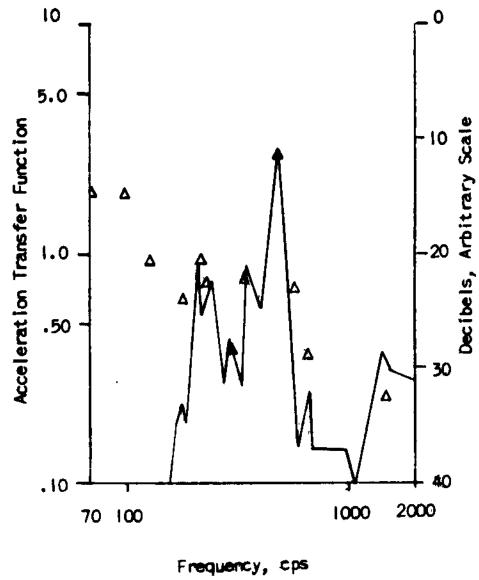


Fig. 5 - Comparison of rms acceleration during random vibration test with transfer function measured during sinusoidal test

throughout the intended mission. In view of the unknown variations in mission environments, some degree of conservatism in these tests is desirable. However, since excess conservatism can be inordinately expensive in design weight and other costs, careful consideration must be given to the degree of conservatism allowed.

It should be noted that there are unavoidable differences between test and flight environments, because the impedance characteristics of the booster are clearly different from those of the electromechanical vibration exciter used during the vibration tests. Thus the vehicle boundary conditions during the tests can differ radically from the actual boundary conditions.

A major concern in the OAO vibration tests was the possibility that the reduced test input motion would not adequately simulate the flight environment in the low frequencies, thereby creating a possible unconservative condition in the tests.

In order to evaluate the degree of conservatism in the OAO vibration test requirements, a comparison was made between anticipated flight levels and measured test levels.

Available data from Atlas launchings indicated that the greatest lateral motion would probably occur at the fundamental bending frequencies of the Atlas-Agena-OAO vehicle at the time of maximum dynamic pressure. The anticipated motion for the upper portion of the vehicle is shown in Fig. 6. Locations of the node lines are of particular significance. It may be noted that the location of the first node line moves upward with increasing frequencies. The structural loading due to this type of motion appears to be highest at 5.1 cps, next highest at 2.5 cps, and then lower for each additional mode of increasing frequency, and as additional node lines occur within the OAO. The motions at 2.5 and 5.1 cps could not be simulated in the test since they are beyond the capability of available test equipment. However, since the motion of the OAO at these frequencies are primarily rigid body, static tests were considered adequate to demonstrate required strength. In the higher modes, the structural loading is a function of the mode shapes, as well as vibration levels. The deformation pattern of the OAO in the 8.3-cps vehicle mode is similar to that of the fundamental 9 cps OAO resonance found during the vibration test; the node line occurring at the base both during test and in flight. Structural loads due to vibration would therefore be maximum at the base, and depend directly upon amplitudes in the forward portion of the Observatory.

A comparison of lateral vibration levels measured at the top of the OAO with anticipated flight levels is shown in Fig. 7. The amplitudes of the data shown are peak values at payload location available from several Atlas launchings; the frequencies were adjusted to coincide with the vehicle bending resonances anticipated during the maximum dynamic pressure condition. It is seen that levels measured during the test were equal to or higher than anticipated flight levels at 9 cps, and also at the other frequencies where the input was reduced to preclude exceeding design strength. Over the remainder of the frequency range shown, the vibration levels were considerably higher than the anticipated flight levels, and therefore, the corresponding shear and bending loads were higher than those anticipated during flight.

A comparison of longitudinal vibration test levels with measured flight data below 100 cps is shown in Fig. 8. It is seen that the peak levels at the top of the OAO during test were considerably higher than the limited flight data available. As in the lateral case, the severity of the structural loading depends upon the mode shape, i.e., upon the amount of mass moving in phase, as well as the vibration levels. A review of the

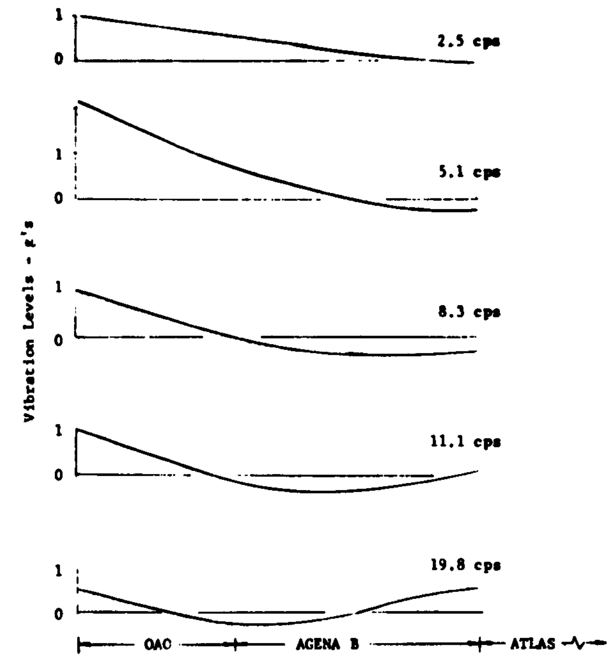
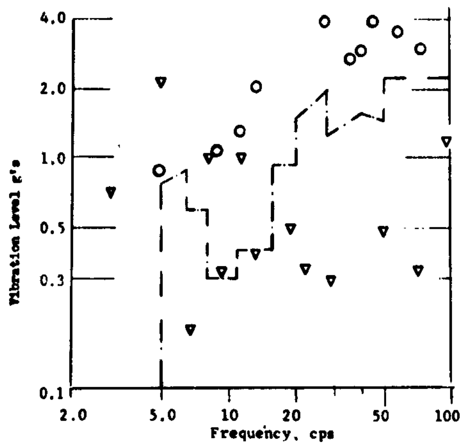
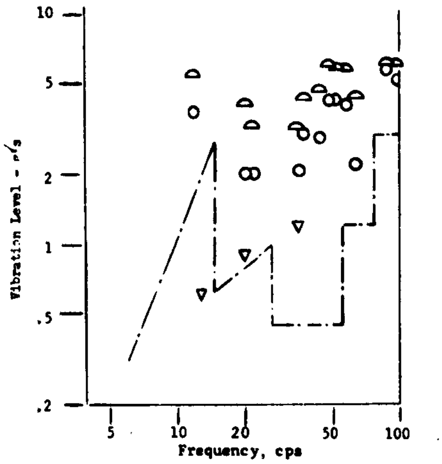


Fig. 6 - Anticipated motion of OAO in low-frequency booster bending modes



—·— Test Data: Input to base of OAO from Shaker
O Test Data: Response at Top of OAO
▽ Flight Data: Forward Payload Locations, Various Atlas-Agena B Shots

Fig. 7 - Comparison of OAO lateral vibration test data with anticipated flight levels



—·— Test Data: Input to base of OAO from Shaker
O Test Data: Response at Top of OAO, Control Tube
⌓ Test Data: Response at Top of OAO Outboard Truss Cap Members
▽ Flight Data: Forward Payload Locations, Various Atlas-Agena B Shots

Fig. 8 - Comparison of OAO longitudinal vibration test data with anticipated flight levels

low-frequency, full-vehicle, longitudinal, mode shapes indicated that more of the OAO mass was moving in phase during full-vehicle resonances than during the tests. However, since the test levels were much higher than the anticipated flight levels, the test loads were considered conservative.

The foregoing discussion is based upon the assumption that the peak response at the top of the full launch vehicle will not change appreciably when the OAO is mounted in place of the payloads used when the measurements were made. This is justified since no appreciable change in the applied force or damping is anticipated, and a brief review of the analysis used showed that the generalized mass would probably remain the same, or increase somewhat. The new full vehicle modes would thus be similar and would still result in maximum motion at the top. Therefore no significant change in response is anticipated.

There is considerable discussion in the published literature on the possibility of large motions occurring if a payload antiresonance coincides with a vehicle resonance. This could occur if the vehicle impedance is considerably higher than the payload impedance, at the payload antiresonance. An investigation described[2] disclosed that this was not the case for the OAO.

In the higher frequencies, i.e., greater than 100 cps, the motion consists of a superposition of complex mode shapes. The exciting forces include aerodynamic turbulence, acoustical excitation, and engine thrust oscillations. It is difficult to estimate the degree to which a motion input vibration test simulates the mission environment in this frequency range. If, as is usually done, the response motion on structural hard points is taken as a criterion, the OAO vibration test will result in a more stringent environment by the amount of the amplification factors measured, since the inputs required at the base were an envelope of the maximum levels measured on past vehicles of similar configuration. An unexpected result of these tests was the amplification from 3 to 4, as experienced in the lateral direction at frequencies between 300 and 600 cps, in which vibration levels of up to 30 g were measured. As a result, the environment in this frequency range is more stringent in the lateral direction than in the longitudinal.

[2]Pulgrano, L. J., Shock, Vibration, and Associated Environments Bulletin No. 31, Part II, (Oct. 1962).

IMPEDANCE CONSIDERATIONS

During the design of the OAO it was expected that just as the OAO impedance could have a significant effect on the motion of the Atlas-Agena, perhaps items of equipment having appreciable impedance might affect the OAO environment. One such item was a star tracker, in which a 5-pound telescope mounted on a beam-like gimbal resonated at a frequency close to the second calculated OAO longitudinal mode. The original OAO analysis indicated that the motion at the star tracker location would peak to approximately 20 gs. During the subsequent OAO vibration tests, wherein all items of equipment were represented by rigid mass mock-ups, the response did peak to about 15 g at a frequency close to that predicted. If the environmental levels for equipment were derived in the usual manner, an input requirement of 15 g to the base of the tracker would have resulted. When the tracker was mounted on a vibration exciter and tested as a component, an amplification of 24 was observed. With a 5-g input, 120 g were measured on the gimbal, producing a reaction force of 600 pounds. Since an input of 15 g would result in extremely high response motion, an impedance analysis[3] was performed to evaluate the probable effect of this high reaction force on the OAO motion. The results of this analysis indicated that at the resonant frequency, the addition of the star tracker would reduce the OAO motion at the mounting location to approximately one-eighth of the original value. However, two additional OAO modes would be generated, one below and the other above the resonant frequency, each with an associated amplification factor of about five.

During the OAO vibration tests, with the star tracker represented by a lumped rigid weight, amplified motion did occur at frequencies near 140 cps. During a portion of the tests, an actual star tracker was installed in place of the rigid weight. The peak vibration level measured on the gimbal in this frequency range was about 20 g rather than the 120 g measured during component tests, as shown in Fig. 9.

This discussion illustrates graphically how overtesting can result from the use of a motion input requirement, i.e., using measured response of a rigid mockup as an input requirement for an actual unit, particularly if an internal dynamical system of the unit is involved.

[3]Shatz, B., and Pulgrano, L., "OAO Equipment Vibrations Including Mechanical Impedance Considerations," Grumman Aircraft Engineering Corporation Report - OAO-TR-DAN 252R-15.0 (Nov. 13, 1961).

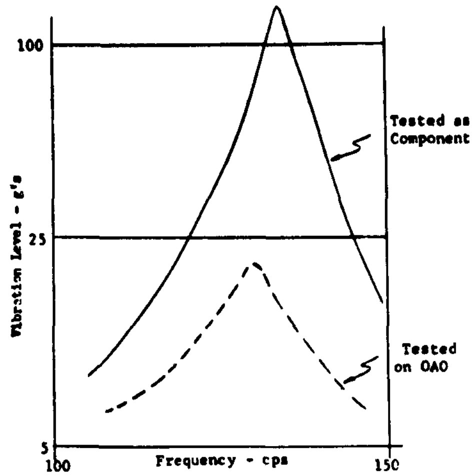


Fig. 9 - Comparison of star tracker motion component test and OAO test internal gimbal response

CONCLUDING REMARKS

Because hindsight is always more reliable than foresight, recommendations for future analyses of this kind are readily presented.

The eight-mass representation, as used in the original analysis of equipment motion, is inadequate without at least the following additional coordinates:

- Longitudinal motion: rotational coordinates for bending in the two lateral planes in addition to the longitudinal translational coordinates.

- Lateral motion: rotational coordinates for torsion and lateral bending, in addition to the lateral translational coordinates.

A lumped parameter representation of the OAO, and the recommended system considered as the minimum requirements for future analyses, are shown in Fig. 10. Additional representation of the experiment package is provided, since it does constitute approximately one-fourth of the weight of the entire Observatory.

In order to incorporate equipment shelf flexibility into the analysis, more than one mass point representing each shelf would be required. In this case three or four should be used, the recommended coordinate system notwithstanding. Although this would produce a system of over 100 degrees of freedom, this complexity is considered justifiable since the amount of time and effort required for such an analysis can be considerably reduced by present day high-speed electronic computers.

The original assumption of 0.04 as a structural damping coefficient still stands as a good estimate, by all standards and experience exclusive of the OAO. In view of the many friction joints providing energy dissipation, a coefficient approaching 0.1 could have been justified. Due

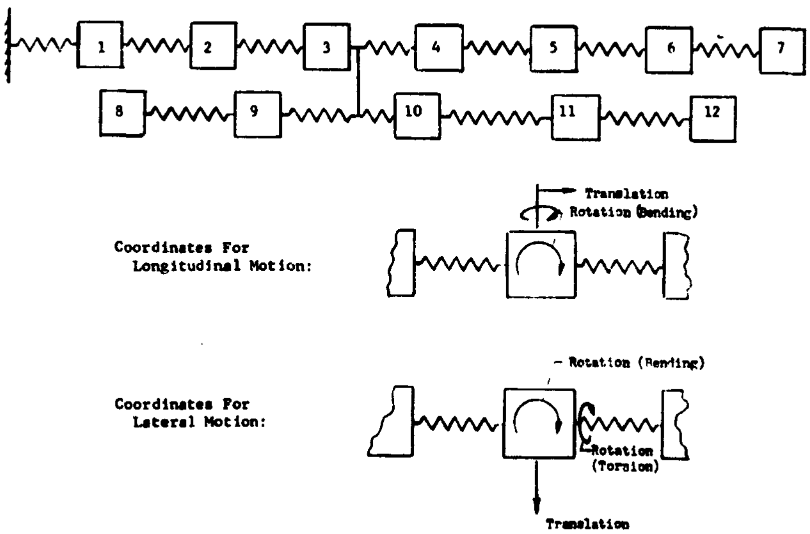


Fig. 10 - Minimum representation and coordinate system for future analysis

to the sensitivity with which the response depends upon this parameter, anything higher than 0.1 could result in unconservative estimates for equipment and structural environment.

In summary, it can be said that the OAO vibration analysis and test program established the following: the predictions of the vibratory environment during the motion input test were subsequently found to be conservative during the actual test program; it was further shown that the motion input test requirements were conservative when compared with the anticipated flight environment. As a result, a high degree of confidence exists in the ability of the satellite vehicle to survive the vibrational rigors of its mission.

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Appendix

RESPONSE OF THE OAO TO VIBRATORY MOTION INPUT

For a lumped-parameter, multi-degree-of-freedom system such as the OAO representation shown in Fig. 2, the equations of motion, ignoring damping can be written (Ref. 1) in a relative coordinate system as:

$$[M]\{\ddot{q}\} + [K]\{q\} = -[M]\{\phi_r\}\ddot{u},$$

where

$[M]$ = mass matrix

$[K]$ = spring matrix

$\{q\}$ = response coordinate relative to input motion u

$\{\phi_r\} = \begin{Bmatrix} 1 \\ 1 \\ \vdots \\ \cdot \\ 1 \end{Bmatrix}$ may be regarded as the mode shape of the rigid body translation, which is required to provide the excitation to the system.

Proceeding in the conventional manner, the normal modes and frequencies are obtained from the free vibrations problem. The forced response is then expressed as a superposition of the normal mode, i.e.,

$$\{q\} = \left[\{\phi_1\}\{\phi_2\} \cdots \{\phi_n\}\right] \begin{Bmatrix} \xi_1 \\ \xi_2 \\ \vdots \\ \xi_n \end{Bmatrix}.$$

The use of the normal coordinates, ξ, uncouple the equations of motion. If this uncoupling is to be retained, any damping that is introduced must be in a modal form. With this, independent expressions are obtained for the normal coordinates, as:

$$\ddot{\xi}_i = \frac{\{\phi_i\}^T[M]\{\phi_r\}}{\{\phi_i\}^T[M]\{\phi_i\}} \times \frac{\ddot{u}}{\frac{\omega_i^2}{\Omega^2}(1 + jg_i) - 1},$$

where

Ω = frequency of input motion

g_i = structural damping coefficient in i^{th} mode.

$j = \sqrt{-1}.$

With the normal coordinates, and the use of proper transformation matrices, shears, bending moments, axial loads, or stress distributions may be easily calculated.

If the system is excited in the i^{th} mode, i.e., if the above expression can be rearranged as

$$g_i = \frac{\{\phi_i\}^T[M]\{\phi_r\}}{\{\phi_i\}^T[M]\{\phi_i\}} \frac{\ddot{u}}{\ddot{\xi}},$$

where j has been deleted since it signifies only phase shift. Since all of the terms may be measured or calculated, the modal damping coefficient may easily be determined.

DISCUSSION

Mr. Davis (GE): I notice your paddles are opened at some time during your flight profile. Could you tell me what the frequency of your open paddles is, and whether the dynamic loads at separation in any way influence your design criteria for the paddles?

Mr. Shatz: The solar paddles? Somewhere around 15 or 18 cps.

Mr. Davis: I'm asking about the open configuration.

Mr. Shatz: Oh, we never did any work on them open. There was no worry about any loads on them during injection into orbit.

* * *

A METHOD FOR SELECTING OPTIMUM SHOCK AND VIBRATION TESTS

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Measurements from tests have shown a tremendous scatter for successive flights at the same pick-up location. The cause of this scatter, ranging over factors such as 4:1, is still unknown and yet designs must be made, tested, and qualified. This data scatter has led to the use of Decision Theory to find a rational solution in the face of uncertainty.

INTRODUCTION

The primary objective for most of the work done on shock and vibration is to obtain equipment which will not fail in service; and the primary tool for achieving this goal is the test machine which produces the severe shock and vibration environments in which these equipments are tested. Although the severity of a test is related to measurements made under service conditions, there are many reasons why a test cannot and should not be a simple repetition of a shock or vibration observed in the field. This paper is concerned with how the various factors involved should be weighed in order to make the best choice of a test level.

First, several of these factors will be discussed in order to make it clear that the uncertainties involved in choosing a test level are both numerous and large. Furthermore, these uncertainties will not be resolved in the near future by such conventional engineering approaches as theoretical analysis, additional data, research, or by using factors of safety and conservative assumptions. This is not so much a technical problem as a problem for "engineering judgment."

It will be argued in this paper that the problem is to arrive at a test level which balances the risk of being too severe and requiring too much redesign against the risk of being too lenient and having too many service failures. It will be shown that a rational method for calculating these risks and weighing the evidence is offered by statistical decision theory [1]. After a description of the principle ideas of this newly-developed branch of game theory, it will be shown how the theory applies to the selection of test levels.

UNCERTAINTIES

Some of the uncertainties in choosing a test are due to unsolved theoretical and technical problems, some are due to uncontrolled variation of factors which influence severity of shock or vibration, some are due to lack of enough data, and some are due to fuzzy definitions of just what a test is expected to accomplish. Since all of these factors affect how good our choice will be, it is not reasonable to spend a lot of time resolving a fine technical point while a far bigger source of error is left untouched. It is for this reason that we shall deviate from the usual technical discussion of an engineering problem in order to explore some other problems that engineers are also expected to solve. Some of the major uncertainties involved in going from field data to a test specification are the following:

- The field data usually shows considerable statistical scatter so that the environment is not uniquely defined. Some scatter might be ascribed to differences in test conditions and some might be due to human error, but the cause is not very clear in most cases. There is some evidence that structures made from the same drawings have markedly different transmissions of high-frequency vibrations [2].

• At the time that a test is needed to simulate the environment of a vehicle, it is often the case that the vehicle is new, and few, if any, measurements have been made. Most data were taken from more-or-less similar vehicles, or the point of measurement was some distance from the point where the equipment was to be installed. Tests are often required for equipment capable of being installed anywhere in a vehicle or in a type of vehicle. We have no theory or measurements showing how to extrapolate data for different structures or for different points on the same structure.

• Measurements of motion alone are not adequate; they should be supplemented by some kind of measure of mechanical impedance. We do not do this now, primarily because it has yet to be shown just what should be measured and how the measurement should be used in designing a test. Our major progress on this problem to date has been to show examples for which impedance effects were significant [3].

• The shock and vibration testing machines and fixtures are quite limited in their ability to simulate many features of an environment such as mechanical impedance [4], details of frequency spectra, simultaneous motion in three directions, and so on.

• The amount of data that we usually have to work with is what a statistician would call a "small sample." But we are not sure that the sample is unbiased nor do we have much idea of the type of statistical distribution from which it was taken. Nevertheless we would like to take 5 or 10 measurements and state a value that will be exceeded, say, once in 1000 times.

• The instructions which we are given defining just what a test is to accomplish are vague and unrealistic. Typical requirements for a test are "to establish confidence in integrity," "to demonstrate the reliability of a design," "to simulate the worst expected environment," "to qualify the design," and so on. If it were possible to measure and reproduce shock severity with reasonable accuracy, and if some upper limit of field conditions could be defined, these instructions would be adequate; but they give no help in finding a good compromise between the risks of being over- or underconservative. Perhaps the shock and vibration engineers have failed to show management that a calculated risk is necessary; but in any event the result is that we do not acknowledge and face those uncertainties.

POSSIBLE ENGINEERING APPROACHES

The ideal way to cope with an uncertainty is to get rid of it. In engineering this is often done by obtaining precise information from research by theoretical analysis, or by gathering more data. All of these approaches would certainly improve our position, but it is doubtful that they will reduce our uncertainties to a tolerable level in the next several years. Theory and research tend to tackle, progressively, more complex problems. To my knowledge there is little current effort to solve, by direct frontal attack, the difficult problems mentioned; they seem well beyond the present state-of-the-art.

When uncertainty is due to statistical fluctuation which cannot be eliminated, the best that can be done is to collect a large quantity of data in order to define the statistical distribution. Again, we could improve our position by this approach, but it is not practical to get the amount of data we would like.

In most engineering solutions to problems there are some residual uncertainties that are dealt with. If an uncertainty seems small enough, it is usually safe to ignore it. For somewhat larger uncertainties, it is common practice to err on the safe side by making conservative assumptions and by using factors of safety. It is to be expected that a design might be lighter or less complex if the uncertainty could have been cleared up, because the penalty paid in the form of overdesign is seldom measured whereas the engineer can get into serious trouble if a design is too weak. However, when the uncertainties get too large, the overdesign becomes more apparent.

If one tries to use conservative assumptions to be sure that a test is to a high-enough level, he will come out with a much more severe test than some field measurement selected at random. He then will be told by designers and management that he has been guilty of overconservatism and that he must find some way to be practical and to cut down his specification. This is the beginning of a frustrating experience familiar to many shock and vibration engineers. As he tries to reduce his uncertainties by more analysis and more data, he makes less headway. In fact, he is likely to become more aware of the magnitude of the gaps in his knowledge. When the engineer doesn't seem to be making progress, the problem is taken over by a supervisor or committee who apply "engineering judgement" to get an "interim specification" which more or less

satisfies all concerned. The interim solution then continues indefinitely because the engineer never finds a good technical solution.

So, in addition to the technical problems of obtaining a good test (or rather because of them), there is still another problem — that of using engineering judgment to find a good balance between the incentives to be on the safe side and the incentives to avoid overdesign. It is this problem which will be examined next in some detail.

ENGINEERING JUDGMENT

We start by asking, just what one means by that very common but nebulous expression "engineering judgment?" Perhaps its meaning is vague because it is best defined in a negative sort of way; it is any means of solving an engineering problem except conventional methods of analysis and experiment such as those taught in engineering school. Good judgment may use any mental tools including ingenuity to take advantage of any special peculiarity of a problem, but an essential ingredient for every situation seems to be a large store of experience with many problems and their solutions. Although details of all this experience may be forgotten, a person with engineering judgment has an admirable talent for guessing the right answer to a problem. He may be unable to explain his decision to others, but it is a good bet that he is right. Unfortunately there is no certain way of knowing when he is going to be wrong, or when an engineer with a strong opinion has good judgment or merely good nerves.

Another ingredient usually present in engineering judgment is that it considers practical as well as technical factors. This aspect is more like business judgment, or even political judgment, than the engineering discussed in technical journals and schools; but such things as cost, schedules, and available facilities as well as weight, reliability, design complexity, and so on are all affected by an engineer's decision. So the overall value of a design is determined by how well the practical factors are balanced against technical knowledge.

It appears that in using engineering judgment a person adds two ingredients to his technical data and mathematical analyses. He gives weight to the predictions of experienced experts, and he considers the practical results of his decision in order to produce the best effect on the overall objectives of a project. It is unlikely that anyone will disagree with these general statements of what we'd like to accomplish; but the process of doing it goes back to the same mysterious intuitive exercise of judgment. It is all too easy for different people to arrive at different judgments and then to have great difficulty in obtaining a meeting of the minds as to which is the better solution.

There would be little point in discussing the unsatisfactory features of engineering judgment if there were no way to improve the process. In recent years, a branch of Game Theory called Statistical Decision Theory has developed to the point where it can be of great help. It provides an orderly method for combining the numerical data from experiments and the advice of experienced technical people with practical considerations and the overall goals of a project in order to arrive at a rational decision which can be explained and defended. The decision is neither infallible nor even the best that might have been devised. It is simply one of the possible decisions suggested for consideration and it is the best of these only in view of the information provided. Better information or more ingenious solutions could still make the decision better. This last difficulty is dealt with by decision theory in a characteristically practical way — by considering whether it is a better decision to seek more information and better solutions or to take action without further ado.

Since decision theory is relatively new and is the basis for the recommendations of this paper, the next section will describe the principle ideas.

STATISTICAL DECISION THEORY

Decision theory provides a technique and a criterion for picking the best course of action from a group of possible actions on the basis of the information available. It does not contradict our common-sense intuitive method of making judgment, but seeks to strengthen it by providing a rational model and methodical steps to a solution. It deals with the same three kinds of information which have always been considered in making decisions under uncertainty by businessmen, investors, military men, oil prospectors, or the man-in-the-street. These three ingredients are:

- The possibilities — the possible actions that we might take and the possible results which might follow our actions.

- The probabilities — the chances or odds of each possible result actually occurring if we initiate the actions.

• The pay-offs — the relative value to us of the different possible results.

In making decisions by judgment, a person has to absorb and consider information of all three kinds until he arrives at an intuitive preference for a particular solution. Since the information is initially distributed among several people, the essential details must be summarized and communicated to the busy decision-maker. Decision theory allows different people to work on the parts of the problem separately and to express their information in a concise and meaningful form. We will consider these three parts of an analysis, in turn. This will lead naturally to a criterion for the best decision.

Possible Actions and Results

A useful way to indicate the various possible actions which might be taken and the results which might follow is a tree diagram (Fig. 1).

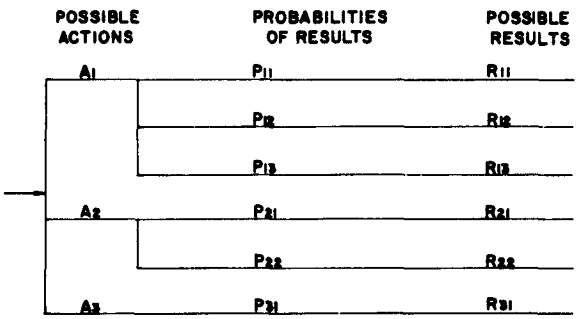

Fig. 1 - Tree diagram of possible actions and possible results

If one starts at the left side of the diagram, he can trace through a succession of branch points to any one of the end results. Each result is reached by only one path. Whether or not future events follow one branch or another will be determined at some of the junctions by our decisions. These alternatives represent our possible actions. At other junctions the course of events are not under our control. Unlike most of game theory, these events are not considered to be influenced by a hostile antagonist. They are simply the result of the operation of physical laws or acts of neutral persons not concerned with helping or hindering us.

The diagram of possibilities is the product of our knowledge of the situation and our ingenuity in devising strategies. There may well be actions and results of which we have not been able to think. This possibility can always be included in the analysis; it might dominate a decision when we are on unfamiliar ground, or it can be negligible in a routine case. In order to keep the analysis from going into more detail than is worthwhile, judgment will be necessary to screen out negligible possibilities.

Probabilities

After the possible results of our actions are described, it is necessary to express how likely it is that an event will actually occur. When a result is determined by some randomizing device such as a spinning pointer (Fig. 2), likelihood that the pointer will stop in the shaded segment is expressed by a numerical probability. Based on the assumption that precautions are taken to assure a "fair" trial, the probability that the pointer will stop over the shaded wedge is simply the ratio of the wedge angle θ to a full circle.

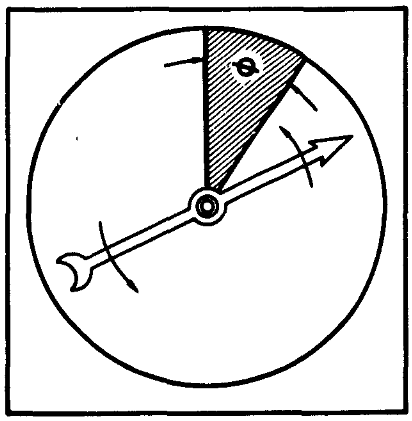

Fig. 2 - Spinner type of chance mechanism

There are many possible events whose likelihood of occurrence is indicated only by a degree of conviction in the mind of an expert. We wish to measure the degree of certainty of the evidence which has caused the expert to expect a possible event. So we ask him first to imagine the wedge angle on a spinner card to be so large that he considers the pointer stopping in the shaded zone to be more likely to occur than the event in question. Then if he imagines the wedge angle to decrease gradually, there will be some angle below which he considers the event to be more likely. So we ask the expert to tell us what probability, P, of the pointer stopping in the shaded zone would seem just as likely to occur as the event being considered. This is a measure of his certainty and is called a subjective probability. In expressing a subjective probability the specialist is simply advising us to rely on the occurrence of a certain possible event no more nor less than we would rely on the spinner. In fact, if the decision is made by assuming that the occurrence of the event will be determined by the spinner with probability P, the decision should not be changed by this substitution.

In practical cases the evidence relating to the likelihood of a possible event may consist of clues from more than one source. Two experts may give different opinions; some test data may be available; an approximate theory may exist, and so on. All of these must be blended into one probability; the means for doing this is Bayes' Theorem. This theorem is proven in texts on probability; it provides for the modification of an "a priori" probability in the light of new evidence to obtain a revised "a posteriori" probability. (The order in which the evidence accumulated is actually of no great consequence; the Latin names are from an earlier more restricted use of the theorem.)

Bayes' Theorem is concerned with how much one learns from experience since it defines how much the probability of a hypothesis increases on the basis of added data. Expressed mathematically, it is

$$P(H) = \frac{P_a(H)\, P(E/H)}{P_a(H)\, P(E/H) + P_a(\bar{H})\, P(E/\bar{H})},$$

where

$P_a(H)$ is the probability of the hypothesis H, prior to occurrence of the event E,

$P(E/H)$ is the probability of E occurring if H is correct,

$P(E/\bar{H})$ is the probability of E occurring if H is not correct,

$P_a(\bar{H})$ is the probability of H being not correct, so it is equal to $1 - P_a(H)$, and

$P(H)$ is the probability of the hypothesis H after the occurrence of E.

Examination of the equation shows that: (1) $P(H)$ will be greater than $P_a(H)$ only if $P(E/H)$ is greater than $P(E/\bar{H})$, (2) the larger $P(E/H)$ is compared to $P(E/\bar{H})$, the more the event E increases $P(H)$, and (3) repeated occurrence of events favorable to a hypothesis make the probability approach closer and closer to 1. Thus, evidence never makes a hypothesis completely certain; but it can make the probability so close to 1 that our choice of action is no different from that for a probability of 1. Further discussion of Bayes' Theorem may be found in Refs. 1, 5, and 6.

Payoffs and Utilities

The goal of our actions is to bring about results for which we have a preference instead of those which we dislike. In practically all engineering, the preferences to be satisfied are not the personal preferences of the engineer. He is hired to act according to the preferences of someone else — stockholders, customers, government, the public, and so on. Thus the value placed on the various possible results depends on the goals of some client.

These values can be expressed by the client, or by someone acting in his behalf, in terms of their utility [7]. The possible results are first listed in order of decreasing preference. Then, in order to assign numerical value to each outcome, they are compared in turn with the best and worst of the results, as in Fig. 3. Figure 3 shows a choice between obtaining result R by Action A_2 or obtaining

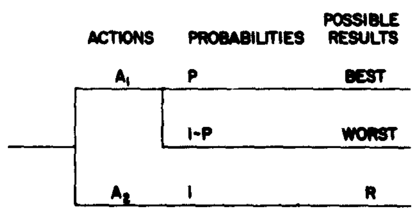

Fig. 3 - Decision tree for measuring utility of result R

either the best or the worst result by Action A_1. There is some probability P of obtaining the best, and $1-P$ of getting the worst result, if Action A_1 is taken. It is apparent that if P is 1, Action A_1 is better; and if P is zero, Action A_2 is better. There should then be some value of P between zero and 1 at which one's preference changes from Action A_2 to A_1. At this value of P, called the utility U of the result R, there is no preference for either prospect. The utility of the best result is always 1 and that of the worst result is always zero.

Note that the expression of preferences has been obtained by asking for a series of simple decisions. It is evident that inability to arrive at a consistent set of values of U, or a sharp value of P at which one's preference changes, will handicap the final decision. But in cases where one action is much superior to another, the decision will not change for wide variations of U.

Although the concept of utility is the best expression of values for use in decision theory, it is not the most natural way to compare different things. Money is the most common means for discussing relative values and may serve as a start toward expressing utility. The concept of trade-offs or exchange rates used in Operations Research are more general and are also very useful.

Criterion for the Best Decision

We have obtained statements from the experts that we may regard various results as being determined by random mechanisms with probabilities that were given. The client has stated the same preference for result R as he has for a chance U of the best result and $1-U$ of the worst. If we then replace each of the results of Fig. 1 by its equivalent chance, we obtain the diagram of Fig. 4. Each path of the modified diagram leads to either the best or the worst results. The probability of reaching the best result through an action can be computed by simple probability theorems. Mathematically the probability that action A_j will lead to the best result is

$$U_j = \sum_i P_{ij} U_{ij}. \tag{1}$$

Since the only alternative to the best result is the worst one, there is no need to compute its probability nor to include the dotted lines on Fig. 4.

Now it is evident that the best action in Fig. 4 would be the one having the highest probability of leading to the best result. Our preference for an action should not change if one possible result is substituted for another of equal value. Hence the best choice in Fig. 1 is the same as in Fig. 4; it is the action having the largest, the maximum "expected utility" defined by Eq. (1).

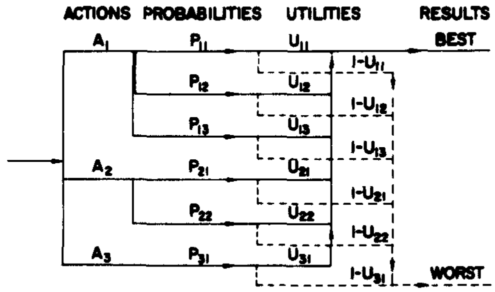

Fig. 4 - Tree diagram with results replaced by prospects of equal utility

Summary of Decision Theory

Although detailed justifications for the steps in decision theory were not possible in the brief review, the major points of the theory were described. These involved description of possibilities, expressing likelihood of results as subjective or objective probabilities, combining evidence of probability by Bayes' Theorem, expressing relative value of results as utilities, and choosing the best action by the criterion of maximum expected utility. The action chosen is simply the one rationally consistent with the smaller decisions made in expressing utilities, subjective probabilities, and possibilities.

APPLICATION TO TESTING

The Decision to be Made

The basic question to be answered by a test is whether or not an equipment design should be changed. This seems pretty obvious, but it is often hidden and distorted by formal requirements of contracts and legally acceptable phrasing. But if there were no question of whether or not a design were good enough, testing would not be done. Since redesigning involves penalties in the form of additional costs, weight, development problems, schedule slips, and so on, as compared to the unchanged design, the decision to change the design is a good one only if

the value of reducing the risk of service failure outweighs the penalties.

Now those of you who have had some experience with present practices will consider an attempt to compare the value of a reduced risk with the penalties of a design change to be wildly impractical because it would require information, which has not been worked out, at nearly every step of the way. Information such as the exchange rate between weight and cost, the penalty for a service failure, the importance of schedule, and so on, is implicit in the judgments made by project managers, but it is seldom discussed with Dynamics engineers. The Dynamics engineers are best qualified to estimate the probability that a part which fails a certain test will fail in service, but they do not now do so explicitly. We do a lot more mental reviewing of risks, possibilities, and consequences than we admit, but it is intuitive and not very systematic.

If we agree that the best design decision would depend upon knowledge of these risks, penalties, exchange rates, and so on, it follows that the probability of a wrong decision increases with each pertinent piece of information that is not weighed in the balance. It seems better to consider these factors, even though our knowledge of them is hazy, than to pretend that they do not matter. An immediate benefit of one's first attempt to work out a testing decision problem is a new perspective on which factors are most important. Some problems which have received a lot of attention in the past do not seem worth working on so long as our major uncertainties remain unresolved. Examples which come to mind are optimizing the analyzer parameters for power spectrum analysis, getting high accuracy in motion pickups, and making different test machines duplicate a test precisely.

One-Level Testing

The most common type of test in use today consists of subjecting the specimen to one level of severity and observing whether the specimen fails. The level is selected before the test, so the potential failure points and the cost of redesigning the specimen are unknown. The decision must be based on an estimated average cost. If a specimen does not fail, the decision to accept the design is automatic. If a specimen fails and it has been decided in advance that this means a redesign, the choice of a test level is illustrated in Fig. 5. As the test level is increased, the penalty for redesigning rises while the penalty for service failure decreases. The total of

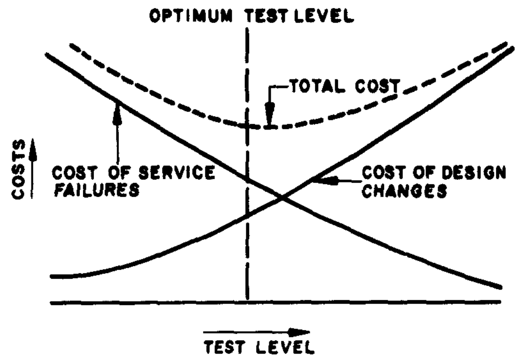

Fig. 5 - Choice of a test level

these costs has a minimum point, which is the optimum test level.

Note that the cost-of-service-failure curve would be steep if the service environment had little variation, if we knew this environment accurately, and if we could duplicate it accurately. In such a case, the optimum test level would be practically independent of the cost of design changes. Our present practice seems to be based on the assumption that we are dealing with this case. Such an assumption needs to be proved, not taken for granted.

It also seems likely that the cost-of-design-changes curve would rise steeply if the test level reached a very high value. There probably exists some state-of-the-art limit of designers' ability to pass tests without major design concessions. At present we have little knowledge of what this limit is.

Except in the two special cases mentioned above, we would expect the optimum test level to correspond to a broad valley in the total-cost curve. This means that moderate errors in the selection or reproduction of a test level make little difference in total cost. The requirement for precise testing stems from legal problems of contracts, not from the design problem.

Testing to Failure

The decision to redesign everything which fails to pass a one-level test is often not automatic. A failure usually starts a debate over the validity of the test, who will pay for the design change, and so on. But a decision to accept a design because of high redesign penalties cannot be made without further testing to find just what test level the design can pass.

If the equipment is tested at successively higher levels until a failure occurs, one can then make a better decision. The costs of redesigning can be more accurately appraised if the point of failure is known. This is a problem for the designer. The penalties (utilities) of such a failure in service can be estimated by management or the customer. The shock and vibration engineer, knowing the test level at failure, is confronted with the problem of stating the probability of the failure recurring in service.

Probability and Utility of Service Failures

If the equipment will be manufactured in quantity and a number, N, of these will be employed in the field, then the possible number of failures might be 0, 1, 2, 3, 4 ----- N. So the problem is to assign a probability and a utility to each one of these numbers. (This is equivalent to stating the probability density of reliability and the utility function, but it seems less abstract.) The redesigned part will be assumed here to have a probability 1 of zero failures, so our interest is in the design that failed at a certain test level. We would like to predict how many of the N equipments will experience an environment more severe than the test. We may need to predict how the strengths will compare with the test article. Both of these judgments will depend on the field data and knowledge of the design, its tolerances, and so on, plus a great deal of other information peculiar to the particular case. Two problems which often arise in analyzing field data will be discussed as examples of the decision theory approach.

The first example is the common one of dealing with data which might be invalid. One can examine the evidence pro and con as to whether the record is a measurement of acceleration or of the tendency for people and their gadgets to make errors. Although we are especially interested in measurements that are high relative to the norm, it may well be that a reading is unusual because of error. Since the data point is either good or bad, there are only two actions which could possibly be right — to throw out the data, or accept it as completely valid. In our desire to have some chance of being "right," we are prone to consider these to be the only rational courses of action, but in using decision theory, one would examine the evidence and express a subjective probability that the data is valid. He would probably find that some course of action between the two "right" ones had a higher expected utility. Sometimes it is better to compromise and to accept being a little wrong than to aim at being exactly right and risk being very wrong.

The second problem is the severe one of taking 10 or 20 measurements of field data and trying to state a shock level which has, say, a 1-in-1000 probability of being exceeded in another measurement. This problem is usually eased by making the assumption that the data is a sample from a normal statistical distribution. The justification for doing this is seldom given and is not a technical one; it appears that something of this sort seems necessary and a normal distribution is the most familiar. I can offer no better motive, and I agree that I prefer this to sketching an envelope across the tops of the data. It is unfortunate that the elegant mathematical treatment of data made possible by assuming a normal distribution tends to lend a misleading aura of certainty to an educated guess.

In dealing recently with a set of 12 measurements, we plotted the points on two kinds of probability graph paper. One had a nonlinear scale such that a normal distribution would plot as a straight line; the other would plot a log-normal distribution as a straight line. The data were extrapolated by using a french curve instead of a straightedge to a level corresponding to an 0.001 probability of being exceeded. The difference between the answers from these equally plausible approaches was startling. In order to get a test that was not completely arbitrary, we used, as a probability of being correct, 2/3 for the normal-distribution method and 1/3 for the log-normal method.

It has been implied here that a test level should have a very low probability of being exceeded in the field. We shall conclude this discussion with an example of an exchange rate between cost and reliability to show how low this probability might be in an expensive missile. Published figures give $10,000,000 as a cost-per-missile for increasing the number of one type of missile deployed. If a change in design were made which increased the probability that each missile would operate successfully, this would be just as desirable as increasing the total number of missiles. Also, the total number could be reduced at a saving in cost without changing the effectiveness of the deployed missiles. The increment of reliability could thus be exchanged for an increment of money. A rough approximation of this exchange rate is obtained by considering a small percent increment of reliability to be of equal value to the same percent increase in number of missiles. This leads to a 0.001 increase in reliability being worth $10,000 per deployed missile.

For hundreds of deployed missiles, this suggests that a design change which produces an increment of reliability too small to be measured with accuracy, is worth more money than most of us will earn in a lifetime.

This answer seems to place a much higher value on reliability than do the reliability people themselves; it is certainly higher than the value implicit in many decisions by project engineers. It would be most inconsistent if we bought reliability under shock at a high price while other engineers were selling it at a much lower price. One possible explanation of this apparent inconsistency is that time also has a very high value in a missile project. An unreliable deployed missile is more valuable than one that is still being designed. It is unfortunate that engineers do not receive a better definition of the exchange rates of reliability, cost, time, weight, and so on, to guide their decisions. Perhaps the cost and time required are too great.

CONCLUSIONS

Because of the many uncertainties involved, having to make a choice of shock and vibration test levels requires that we take a calculated risk. The optimum test decision compares the risk and penalties of service failures if a design is accepted, against the monetary, weight, schedule and development penalties if a design is changed. A good decision requires statements of the probability of failure of equipment in service as a function of the level at which the equipment failed in a test, statements of the penalties for redesigning the equipment, and statements of the penalties for failure of the equipment in service.

These statements should come from the environmental engineer, project design staff, and the customer, respectively. Statistical decision theory is recommended as the most rational method of selecting the best course of action.

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DISCUSSION

Dr. Curtis (Hughes AC): Ralph, during your paper I did not hear any mention of the Monte Carlo method and yet what you are saying seems very analogous to this procedure. Can you point out the similarities and differences?

Mr. Blake: I don't think I can because I'm not that familiar with the Monte Carlo method. I suspect it is not too closely related to this for that reason.

Dr. Curtis: I think I can describe it even though I don't understand it. I guess the basic difference is that you do not assign a single probability to an event, but, hopefully, you know the probability distribution of an event happening. You then assign this to the various things which may go on during the process and you go through the game and play it many times. In effect, you spin your roulette wheel as many times as necessary and, in that way, arrive at the most probable final result, based solely on playing the game a significant number of times. This then leads you to an optimum choice of actions to get the most probable or the most

likely result. I have been groping to find the difference between this and what you have been describing.

Mr. Blake: Well, I had thought that the Monte Carlo method was a means of using a computer as a sort of a random model to run repeated experiments and to find statistical distributions of results. I haven't seen the Monte Carlo method suggested as a reference in any of the papers or books that I have read on decision theory, so I suspect that they are somewhat far apart.

Dr. Curtis: I think they have been used mostly for reliability prediction techniques; however, I see no reason why, instead of using, say, a plain time-between-failure-criterion, we couldn't use an environmental-test-level or an environmental-field-level for this same variable, if we had the distribution of field environment.

Mr. Blake: The probabilities used in the Monte Carlo method, I suspect, are the so-called true types of probability. Many statisticians are not very fond of the notion of subjective probabilities as yet.

Dr. Curtis: I think, if I recall correctly, you have a fair amount of freedom as to the kind of distributions which may be used. They may be based on experimental data or, if you don't know this, you can assume a distribution. I guess the calculations are easier if they are Gaussian but there is no restriction, since one has a digital computer, to any particular kind of favorite distribution.

* * *

THE USE OF MEASURED DATA FOR ESTABLISHING HARDWARE DESIGN CRITERIA FOR PRIMARY GROUND SUPPORT STRUCTURES

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The Martin Company, Denver Division

This paper points out how measured data for full scale structures was applied to rewriting existing design and test criteria, and also how the data was utilized in preparing new structural design criteria for possible follow-on programs.

INTRODUCTION

The primary technical requirements for establishing structural design criteria through the use of measured data, comprise dynamic effects, such as damping, stiffness, and coupling effects, rather than static effects which are usually determinable and transferable to design criteria. This paper points out how measured data for full scale structures was applied to the rewriting of existing design and test criteria, and also how the data was used in preparing new structural design criteria for possible follow-on programs.

Several interesting topics are discussed under launch stand criteria. It is shown how the total spring constants at the TITAN I battleship tank attachment points were measured, and how these measured values were used with theoretical studies to predict missile-stand response to gimballing checkout procedures. The measured spring constants were used to rewrite the stiffness requirement specification for later stands, and extrapolation yielded stiffness requirements for the TITAN II operational launch stands. It is shown also how data taken during abortive and malfunction captive firings were used to establish maximum loads to be applied to the stand. Simple handshake tests revealed data regarding missile-stand combination damping and coupling with the erectors and umbilical towers, and these data helped establish safe stability criteria for full duration run captive firings. Other stand data used for establishing future design guides involved torsion tests for roll instrumentation calibration, analysis of data which showed the effective flame impingement pressures during missile lift off, and measurements of missile-stand response caused by wind-induced oscillations.

Tests performed on the missile erector tower produced results that can be used in writing dynamic loads criteria for erector modifications for follow-on programs. It is shown how vibration was induced and how the response compared with theoretical calculations. The loads induced as a result of sudden stop during erection are discussed in detail, since they may govern the design of many structural members.

Natural frequency measurements for the complete umbilical tower were made in order to be sure all attached masses were properly considered. The results of these measurements were combined with theoretical calculations to establish criteria which specified upper limits upon the permissible boom rotation velocities that could occur on later R&D towers.

Numerous tests were made on the missile transtainer in order to determine maximum load factors that would be applied to the missile during handling and road transportation. It is shown how theoretical and experimental results for TITAN I are combined to develop design criteria for the TITAN II transtainers. It is interesting to note that the missile can be mounted close to the node points of the free-free fundamental transtainer frequency, thereby eliminating somewhat the transference of transtainer vibrations into the missile.

In order to establish the final design criteria for the operational TITAN II Shock Isolation System (SIS), numerous tests on full scale breadboard components were made, and tests

on a full scale system were also carried out. It was necessary to establish the true damping of the components and complete SIS, which incorporated Coulomb damping. The operational equipment must have accurate damping since the damping affects the residual after-blast-deflection of the IMU prism relative to the autocollimator beam, reaction time after sustaining a hit, lock-up strut stick out, maximum missile loads, and overturning stability. The tests revealed valuable data early in the program and, therefore, allowed changes to be made in the operational hardware without field retrofit; consequently, large economic savings are resulting.

LAUNCH STAND

During captive firings of missiles, dynamic loads are imposed on the missile and stand due to engine thrust build-up, malfunction of one or more thrust chambers, engine gimballing, and engine shutdown. To prevent structural failure of either the missile or stand, it is necessary to determine the dynamic loading which might be imposed on the missile and stand. The test stands for the TITAN I missile were designed to accommodate both stages in a side-by-side configuration. The idealized dynamic model of the stand is the same for both stages and is shown in Fig. 1. The stiffness and mass parameters of each stand are different. The spring constants for the model were determined experimentally from a series of static load tests. Vertical loads, moments, and lateral loads were applied at the missile attachment points (top of A-frames) by hydraulic jacks. The resulting deflections at points of interest in the stand were measured by mechanical dial gages mounted to a built-up ground structure. Figure 2 shows a photograph of the test setup to take these measurements. By applying the loads and moments in steps or increments, load deflection curves, which established the spring constants for the model of Fig. 1, could be plotted. Total spring constants from the top of the A-frames to the ground (Table 1) were found by summing all the spring constants in series. These measured spring constants were used in the calculation of frequencies and mode shapes of the missile-stand combination.

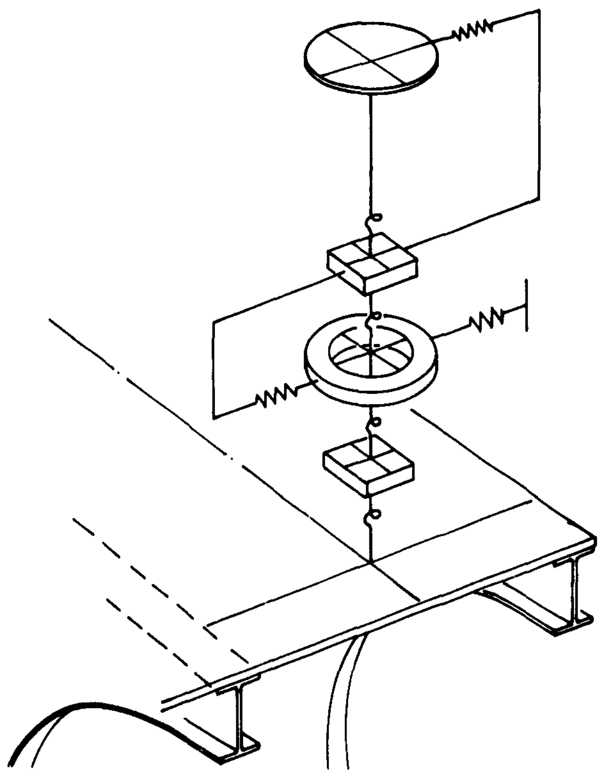


Fig. 1 - Isometric dynamic model of Denver stand

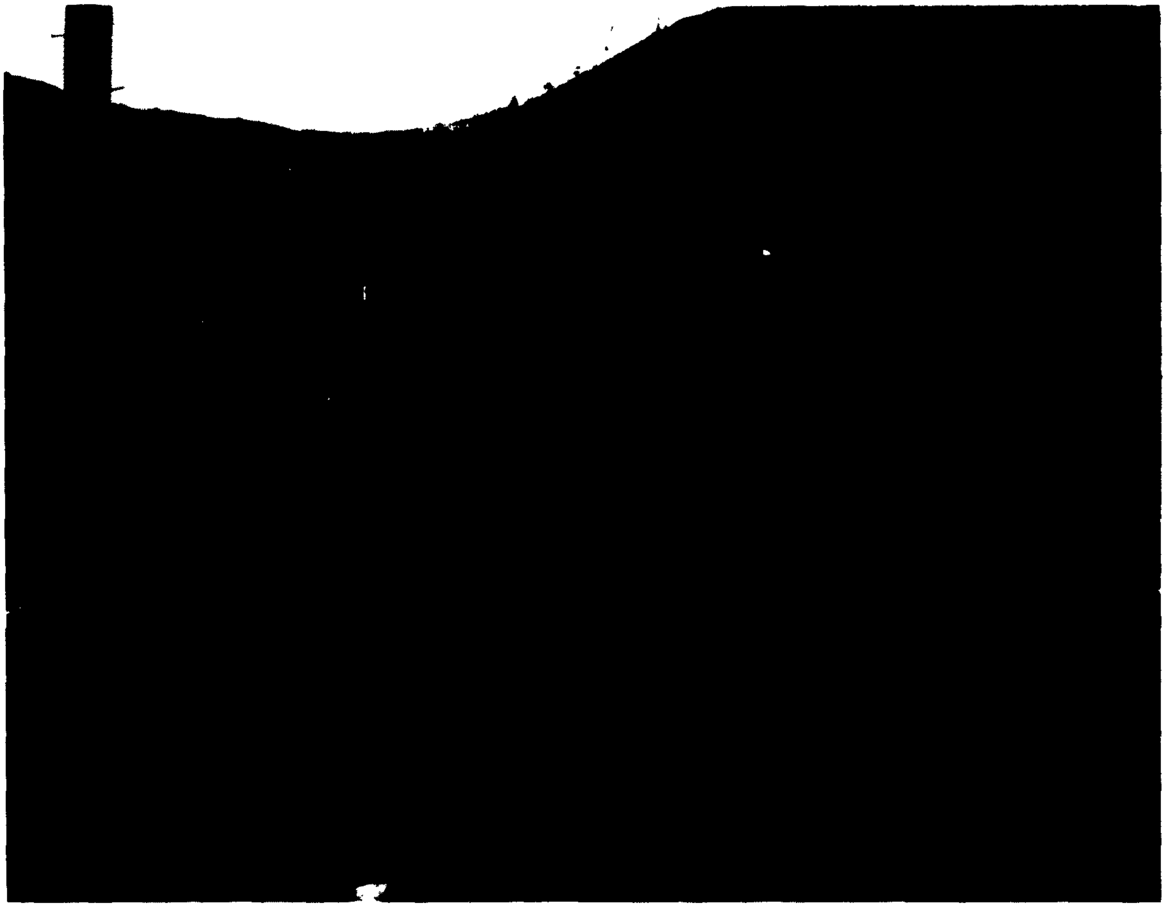


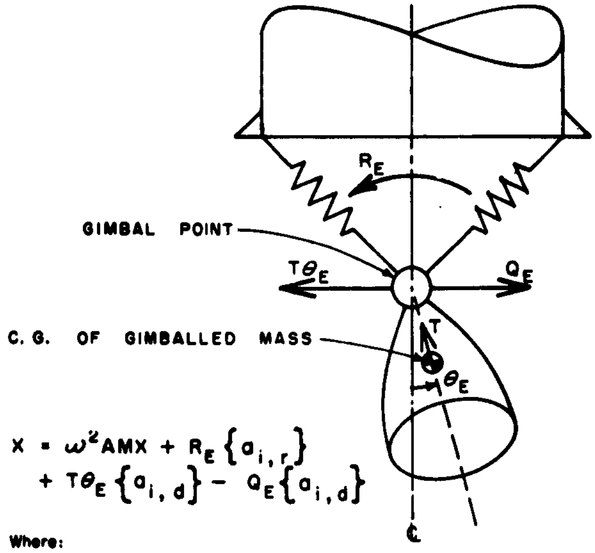
Fig. 2 - Test stand and arrangements for static load tests

TABLE 1
Total Spring Constants

Stage	Total Spring Constants		
	Vertical (lb/in.)	Horizontal (lb/in.)	Rotational (in. lb/rad.)
I	1.74×10^6	0.21×10^6	5.90×10^9
II	0.64×10^6	0.34×10^6	2.70×10^6

The test articles in the early R&D TITAN Program were Stage I and Stage II battleship tanks. This name evolved from the fact that a thick steel plate was used for the skin as compared with a thin aluminum skin for the missile. In this way, repeated tests could be made on the same test articles. Except for the skin, the battleship tanks were identical to the missile itself.

In order to checkout the airborne flight control system prior to a static test firing, it was necessary to gimbal the engines at various frequencies. The gimbal mechanism was controlled by a hydraulic actuator which in turn was actuated by an autopilot. For a given frequency and gimbal angle, the engine inertia forces caused a shear and moment which excited the battleship tank-stand system. See Fig. 3 for the mathematical model and matric formulation. When the engine gimbal frequency was near or at a resonant frequency of the battleship tank-stand, maximum excitation occurred. The response of the system was found for a range of frequencies with a 1-degree engine gimbal angle input. Limits were established for the magnitude of displacements and moments so that no structural damage would occur to the stand or battleship tank. A plot of frequency versus permissible engine gimbal angle was thus established to allow complete flight controls checkout. These



$$X = \omega^2 AMX + R_E\left\{a_{i,r}\right\} + T\theta_E\left\{a_{i,d}\right\} - Q_E\left\{a_{i,d}\right\}$$

Where:

X is the column matrix of displacements at each station.

ω is the circular forcing frequency, a scalar.

A is the enfluence coefficient matrix.

M is the mass matrix.

R_E is the moment due to engine rotation, a scalar.

Q_E is the shear due to gimballed mass translation, a scalar.

T is the engine thrust, a scalar.

θ_E is the angular displacement of the thrust chamber, a scalar.

$a_{i,r}$ is a column matrix of displacements at the i^{th} station due to a unit moment at the gimbal point.

$a_{i,d}$ is a column matrix of displacements at the i^{th} station due to a unit shear load at the gimbal point.

Fig. 3 - Response to forced-engine gimballing

results also helped establish the test requirements for later firings of actual missiles.

Two TITAN I captive hot firing tests resulted in serious malfunctions which made it necessary to shut down the engines a few seconds after ignition. Valuable information was obtained from these malfunction firings regarding dynamic loads imposed upon the missile-stand combination.

The first malfunction occurred with a tandem missile which had the second stage full of water. One engine fired up to full thrust while the other engine achieved only partial thrust; then one engine lagged the other in shut down such that at one instant there was almost 75 percent difference in thrust between the two engines. This thrust difference, of course, induced severe dynamic loads upon the missile-stand combination to the extent that missile bending moments in some locations were equal to around 70 percent of ultimate loads. The second serious malfunction occurred during the captive firing of Stage I alone. In this firing, the No. 1 engine swung to the inside and went hard over, while the No. 2 engine stayed aligned and built up to full thrust of 150,000 lb. An automatic kill resulted. Loads and moments from the load cells, shown in Fig. 4, were recorded during the abortive firing. The results are shown in Fig. 5. Note that engine kill occurred approximately 4 seconds after ignition signal, and that considerable vibration resulted. The average F_x reading was 8500 lb during firing. The thrust chamber pressure for the No. 1 engine indicated a thrust of approximately 42,000 lb and the thrust chamber pressure for the No. 2 engine indicated a thrust of approximately 150,000 lb. Since the No. 1 engine swung to the inside, it had a net offset from the pure vertical of approximately 2.5 degrees.

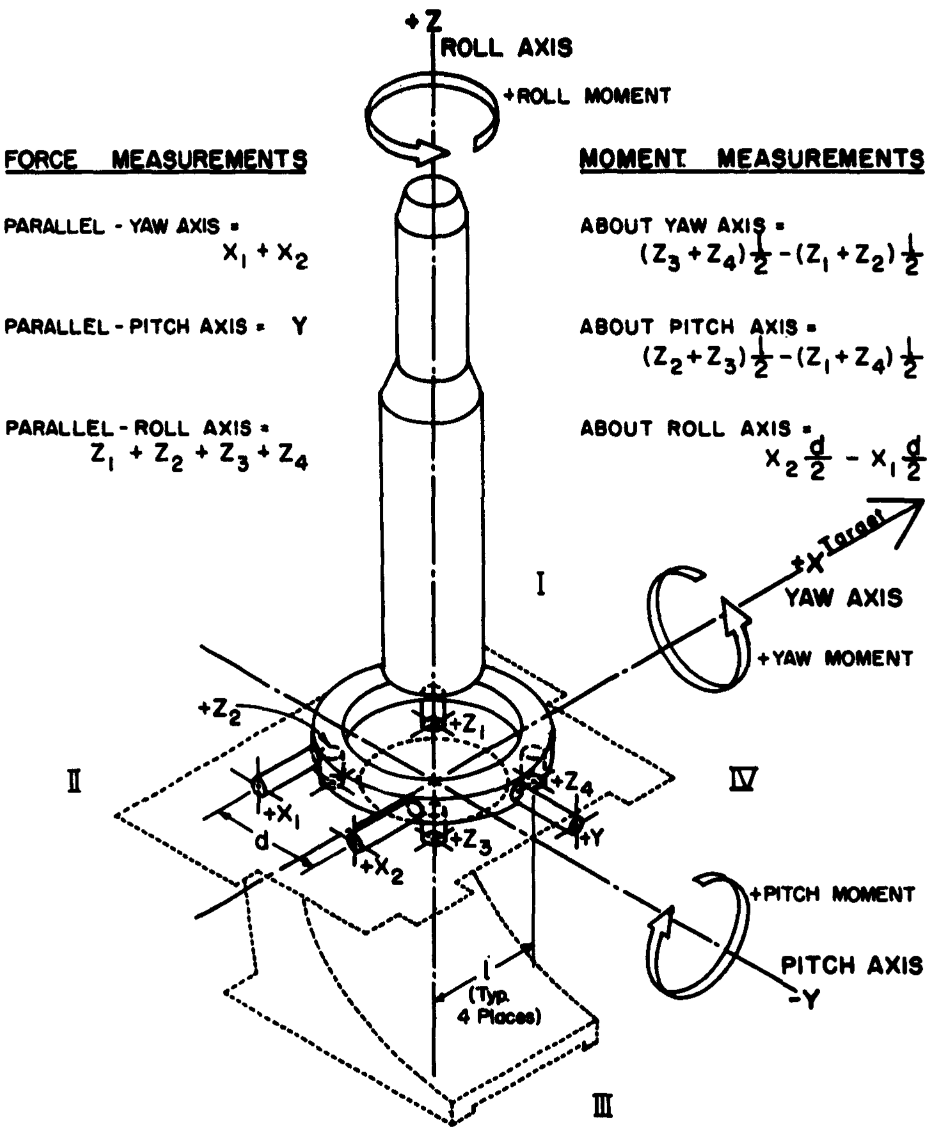


Fig. 4 - Load cell system for TITAN test stands

(This engine was originally canted out from the true vertical by approximately 2.5 degrees; thus, (150,000 + 42,000) sin 2.5 deg = 8400 lb). This calculated 8400 lb agreed very well with the measured 8500 lb on Fig. 5; therefore, there was little doubt that the No. 1 engine went hard over to the inside while the No. 2 engine was approximately at full thrust. Missile moments and shears imposed by this unusual firing condition were back-calculated from the observed data on Fig. 5. This calculation indicated that the maximum moment received by Stage I at the missile attachment points was 3,140,000 in.-lb. This value was less than half the ultimate capacity of the missile. Maximum shears induced upon the missile were between 1/3 and 1/2 of the ultimate missile capacity.

After comparing theoretical studies with the measured data from the two malfunction firings, it was observed that for a tandem full missile, certain combinations of one engine firing and shutting down, and then the other engine firing could lead to destructive, dynamically-induced, missile bending moments. Although such malfunction firings were not very probable,

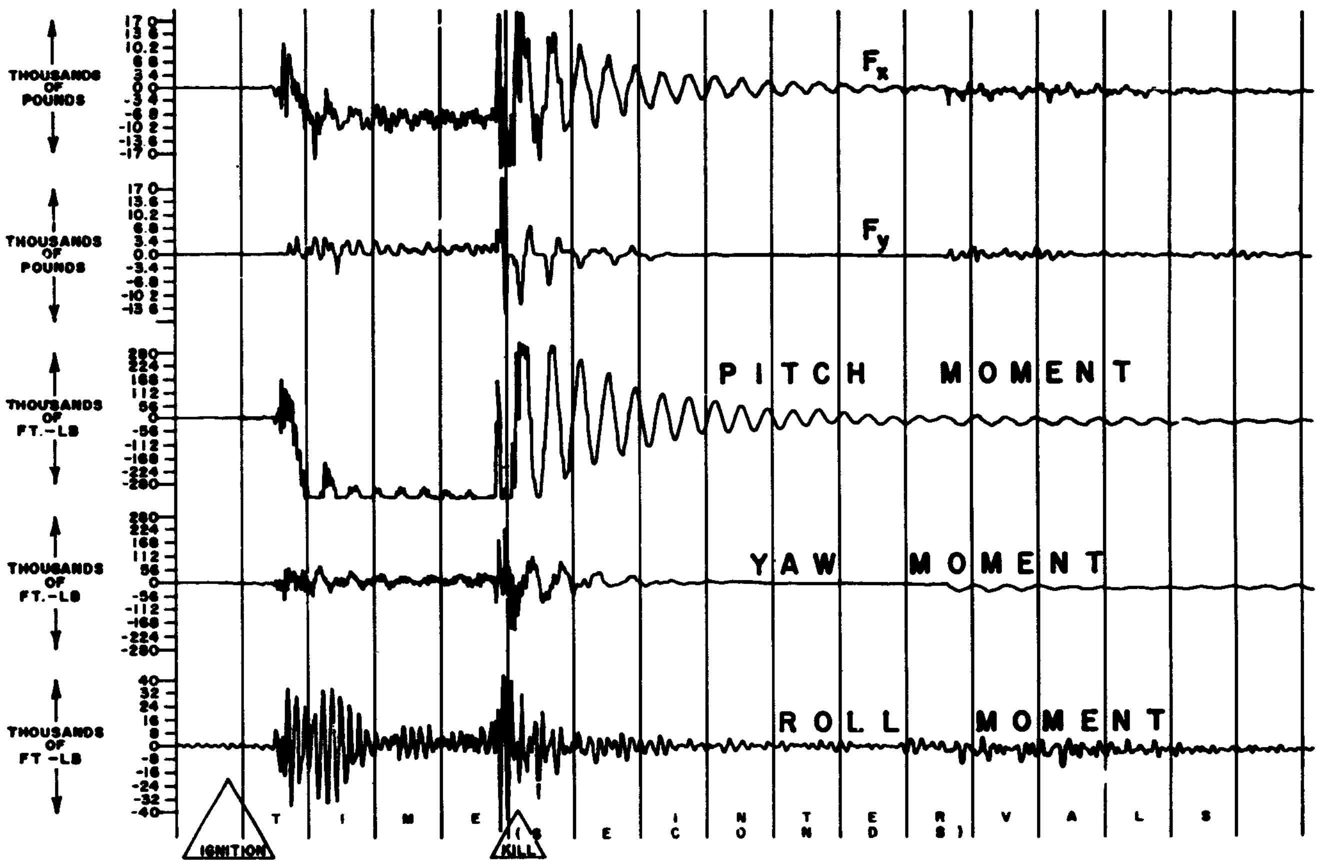


Fig. 5 - Missile abortive firing data

it was decided to increase the stand stiffnesses for the follow-on operational TITAN II stands. The horizontal stiffness was increased by a factor of 3, and the rotational spring constants were increased over 40 percent. As a result, it was felt that TITAN II abortive captive firings could never induce moments and shears which could cause the missile to fail.

The vertical and horizontal load cell (Fig. 4) kill parameters were very important to captive test firings. A build-up of dynamic loads which could cause a structural failure in either the stand or missile could be prevented by not exceeding these parameters. Therefore, it was very important that the load cells functioned properly for all captive test firings.

A simple test was devised by which the calibration and functioning of the load cells could be checked. When the missile is vibrating in its fundamental mode on the test stand, the bending moment and horizontal shearing force at the load cells can be very accurately calculated for a given tip deflection of the missile. The fundamental frequency was low enough that the missile could have been excited easily by pushing on it by hand. The amplitude of vibration was determined by attaching a displacement indicator to the tip of the missile. By manually pushing the missile at varying tip amplitudes and reading the corresponding load cell readings, the calibration and proper functioning of the load cells could be checked very quickly and simply.

This simple test for checking the load cells led to several other tests that could be made easily without any other instrumentation or special test equipment.

The response of the rate gyro could be checked easily since, by knowing the mode shape and frequency, the rate of change of slope ($\dot{\theta}$) at the rate gyro missile station could be accurately calculated for a given missile tip amplitude.

Damping tests for the missile-stand combination also could be run easily. The missile was manually excited to various tip amplitudes and the vibration decay curves were recorded by the load cell traces. By measuring the amplitudes of the decay curve, the amount of damping present in the missile stand combination was determined. The results of one of these damping tests are shown in Fig. 6.

The manual vibration tests revealed that the fundamental frequency varied slightly with the displacement of the tip; i.e., the frequency was a function of tip displacements. For the particular TITAN I missile on the stand, an average of several measurements indicated that this effect could be approximately expressed as:

$$f = f_o - 0.08\ x_{tip},$$

where

f = frequency when tip is displaced x_{tip}, and

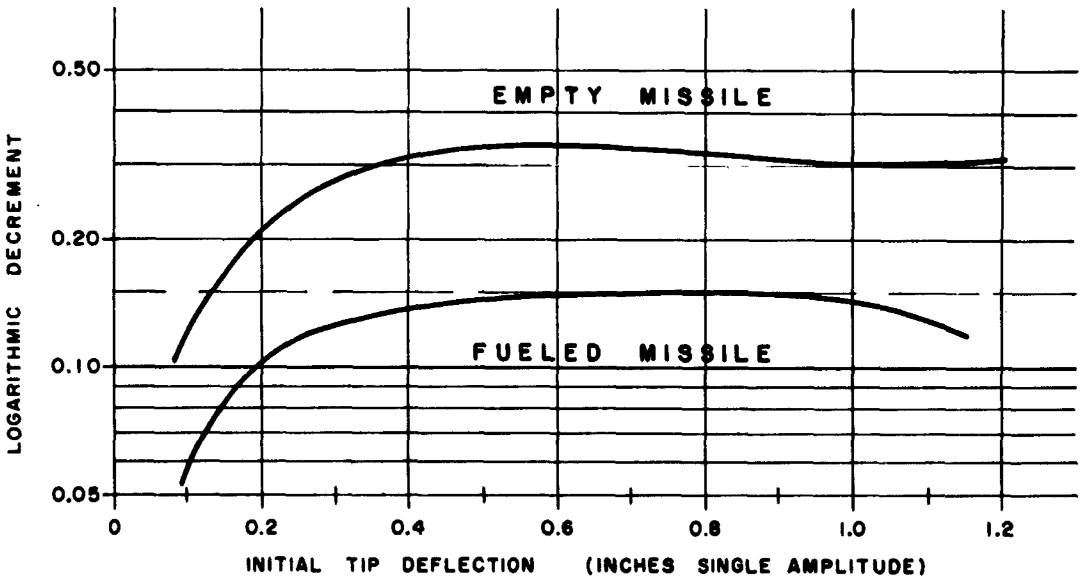


Fig. 6 - One-cycle logarithmic decrement; Lot B missile on stand P-19

f_o = frequency for very small tip displacement.

This observation permitted hot and cold gimballing to be planned into criteria for later missiles in such a manner that sinusoidal gimballing, fairly close to a natural frequency, could be accomplished since, even if gimballing at the natural frequency was started, the natural frequency would change with displacements and therefore help prevent excessive amplitudes from developing.

The shake tests also revealed some interesting torsional frequency information. Cross coupling caused the missile to vibrate torsionally and it was observed that a full missile sometimes had somewhat lower torsional natural frequencies than an empty missile. Normally, one does not assume that liquid propellants affect these frequencies, however, it appears that some fluid, by getting trapped between stringers and possibly through skin friction, is having an effect.

The first captive firings of the TITAN I missile induced high roll readings on the load cells shown in Fig. 4. It was first assumed that the engines were misaligned in roll; this could be catastrophic at launch. This roll moment persisted even after the engine actuator adjustments had been checked many times. It was difficult to determine just where the unknown forces, causing roll moment, were originating. The torus ring just below the engines and above the load cells was loaded tangentially and the roll moment was recalibrated. The system was good. It was finally decided that the sudden release of load might cause the thrust-mount torus ring to rotate. A missile was loaded and unloaded and, as load was changed, a slight roll occurred. It was discovered that a very slight misalignment of the vertical load cells could cause this roll because of the long lever arms. The alignment was improved and the specification requirements were changed for future stands so that the load cells could measure any missile-induced true roll that might occur.

During the first flights from Cape Canaveral the flame shield on the torus ring (Fig. 7) was blasted loose; this required an improvement in design. It was difficult to predict maximum loads, so measured data taken by the vertical load cells (Fig. 4) were used. The load cells recorded the total integrated flame pressure on the torus ring as vertical load versus time while the missile lifted off the pad. An appropriate safety factor was applied and the design of the new flame shielding was rapidly completed.

Fig. 7 - Hold-down arms and flame shield on torus ring

Before the first launch of TITAN I, it was necessary to predict accurately the response of the missile to wind-induced oscillations. Several methods of analysis were available for smoke stacks, but the only treatment of the subject for missiles that was available was being developed specifically for TITAN. This work was later expanded and published.[1] It was desired to obtain some experimental checks on the computations, since the theory was relatively new and several assumptions had to be made that involved basic judgment. There were steady winds of 20 and 38 mph on the two days before the first planned flight. The load cells shown in Fig. 4 were turned on and the maximum moments measured. The missile vibrated randomly as predicted and the primary motion was perpendicular to the wind direction. The resultant moments at the missile base were determined, plotted, and compared with the theoretical predictions as shown in Fig. 8. The agreement was very good, and this full scale measured data enhanced confidence in the theory so that future predictions could be made for other nose cone shapes and for TITAN II.

Another interesting observation was made during the first captive firings regarding longitudinally-induced dynamic loads which resulted from the engine start transients and shutdown. With the missile stand structure idealized as a long string of masses interconnected by springs, the engine thrust build-up-versus-time suddenly applied, and the responses calculated, the dynamic loads appeared considerably higher than the measured values. It was observed that the system was cross-coupled and that during start-up of the engines the structure went into lateral and torsional oscillations as well as longitudinal. Therefore, some of the initial available energy was distributed into the lower lateral modes of the structure and resulted in more lateral vibration. Thus, it was decided to construct more realistic dynamic analytical models in which cross coupling was considered. These types of models were used to predict, more accurately, longitudinal dynamic loads for later TITAN models.

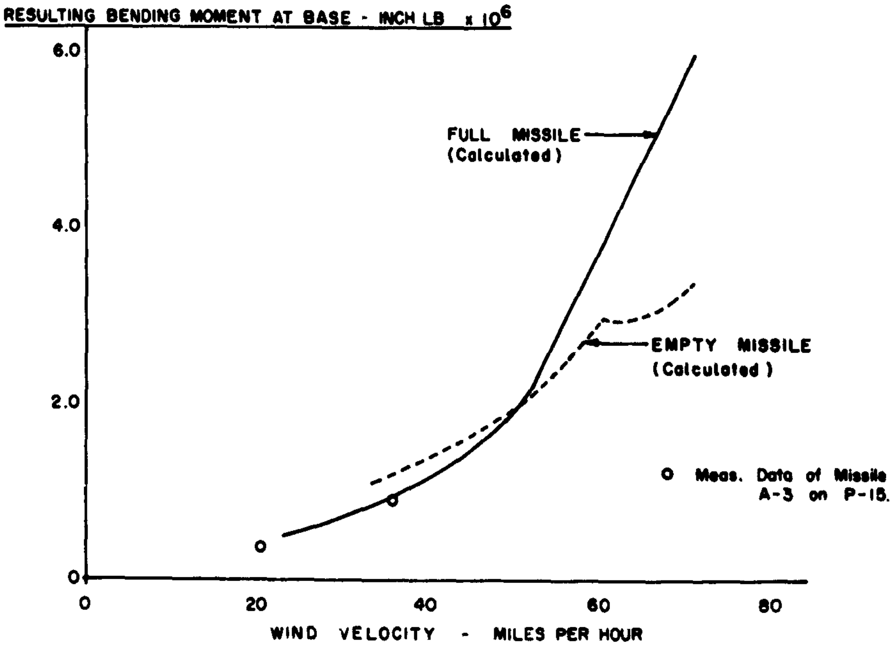


Fig. 8 - Wind-induced oscillations

ERECTOR TOWER

Vibrations of the missile erector tower in the pitch direction presented a rather unusual tower problem. A cross section of the erector

[1] Ezra, A. A., and Birnbaum, S., "Design Criteria for Space Vehicles to Resist Wind Induced Oscillations," American Rocket Society Journal, 1764-1766 (Dec. 1961).

tower will reveal it to be "U" shaped. Therefore, when the erector is deflected in the pitch direction, it will also twist about the shear center. The dynamic model for the coupled erector tower-stand-missile is shown in Fig. 9. The motion of the erector tower is coupled to the missile through the "C" frames on the test stand. Machine solutions were run for the coupled vibrations problem of the missile and erector tower. From the machine solutions, the bending moment and shear loads at the base of the erector tower were calculated for given tip deflections. Strain gages (from which bending moment and shear loads could be determined), were then installed at the base of the erector. A simple test was then run in the following manner. A scale was attached to the top of the erector and a surveying transit was set up on the ground to sight on the scale. Several men then went to the top of the erector and, by shifting their weight back and forth, caused the erector to vibrate in the fundamental mode. The transit man then called out the tip amplitudes of the erector (over the intercom system) to the personnel running the recorders, who then marked the corresponding tip amplitude on the strain gage recordings. These data, when compared with the theoretical machine solutions, reflected very good agreement. The measured fundamental erector frequency in the pitch plane was 1.40 cps as compared with a calculated value of 1.378 cps; the measured fundamental frequency in the yaw plane was 1.87 cps as compared with a calculated value of 1.940 cps.

It was desired to obtain accurate, measured, frequency data in order to be sure that engine gimballing would not be programmed at the erector-stand-missile fundamental frequency. Also, accurate erector mass and stiffness data were needed in order to compute maximum dynamic loads on the erector while it was being lowered or raised. The good agreement of measured frequencies with calculated values increased confidence in the analytical dynamic model. Figure 10 shows the analytical model that was used to compute maximum erector loads when a sudden stop occurred while lowering. The method of analysis is outlined in Ref. 2, and Fig. 10 gives the formulas for computing the maximum deflections that occur at the top of the erector as a result of a sudden stop. The deflections and loads at all mass stations may be computed as shown in Ref. 2. Both stage erectors took a minimum of 3 minutes to raise or lower. When stopping just before reaching the horizontal, for the first stage erector, analysis showed that the dynamic deflections were about 15 percent of the static deflections, while for the second stage erector, they were about 38 percent of the static deflections. Total deflections at the top were 8.2 inches for the first stage and 0.7 inch for the second stage. Total member loads were computed and found to be safe; however, studies for TITAN II and DYNASOAR indicated that some erector members would have to be strengthened. These requirements were included in early specifications for the DYNASOA C vehicle.

UMBILICAL TOWER

Full scale measured data were obtained just after the umbilical tower was completed in order to verify initial design criteria regarding dynamic loads, which were applied to the tower by boom rotations and wind. The fundamental flexural and torsional frequencies were measured by shaking the completed tower, and the measured results agreed well with calculated values; consequently, it was felt that the mass and stiffness data that were used in the analytical dynamic model were reasonably accurate. In general, the impulse from the boom rotation was assumed to excite the first three torsional modes concurrently with wind gusts, which excited basically the lower bending modes. The analysis indicated that the tower members had sufficient cross-sectional area, provided the cable ducts were attached to the main chord members instead of being attached to the diagonal members, in such a way as to cause combined lateral and axial loads on these members.

The method of computing flexural frequencies was changed slightly. In the usual technique, influence coefficients would be computed for the indeterminate structure with a load at each panel point. In order to save time, the tower was assumed to act as a long beam-like structure of variable EI and shear area. The chords were assumed to carry the bending and the diagonals the shear. By studying exact solutions for bay truss deflections, it was discovered that the shear deflections between bays could be approximated as

$$\Delta_s = \frac{3P\ell_i}{A_i E}\,. \qquad (1)$$

where

- P = average shear across section,
- ℓ_i = vertical distance between panel points,
- A_i = effective shear area, a single diagonal for X and K trusses, and
- E = modulus of elasticity.

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[2]Cox, H. L., "Response of Missile Erector Towers to Sudden Stops or Impact Loads," J. Roy. Aeronaut. Soc., 61:694-696 (Oct. 1957).

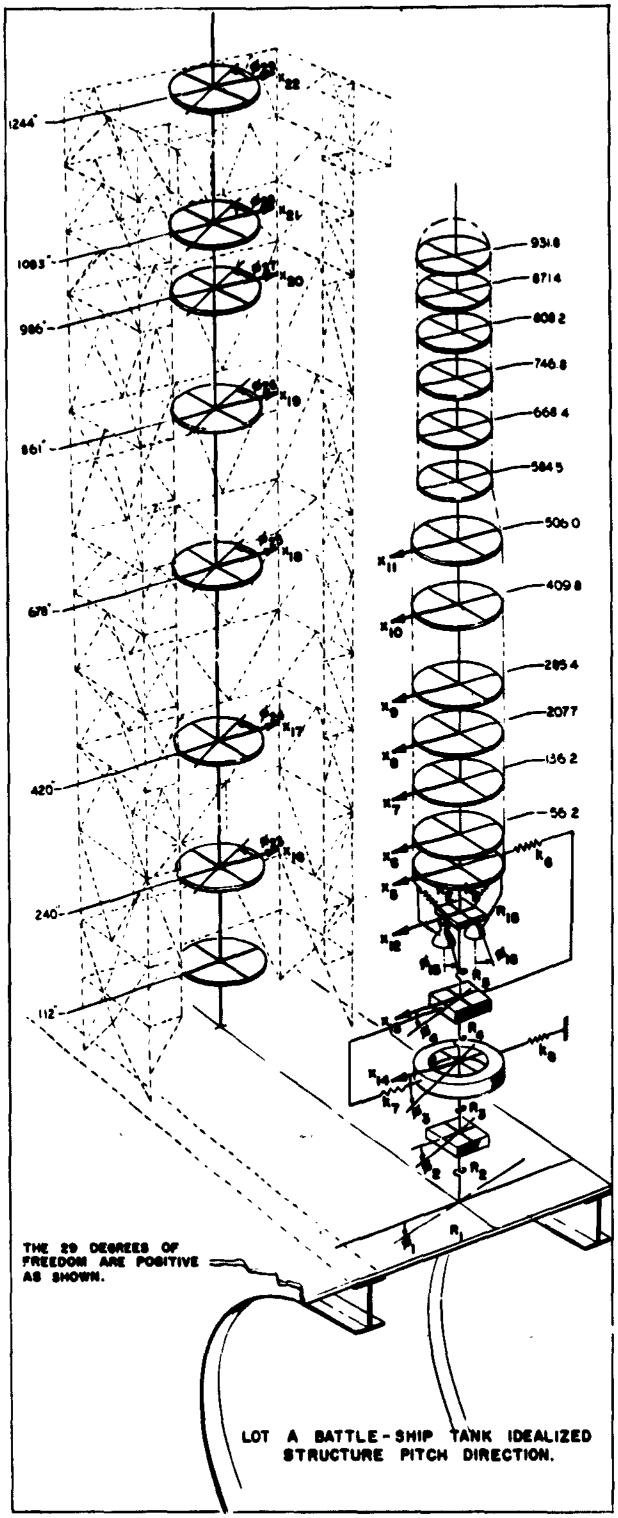

Fig. 9 - Dynamic model for coupled erecta tower-stand-missile

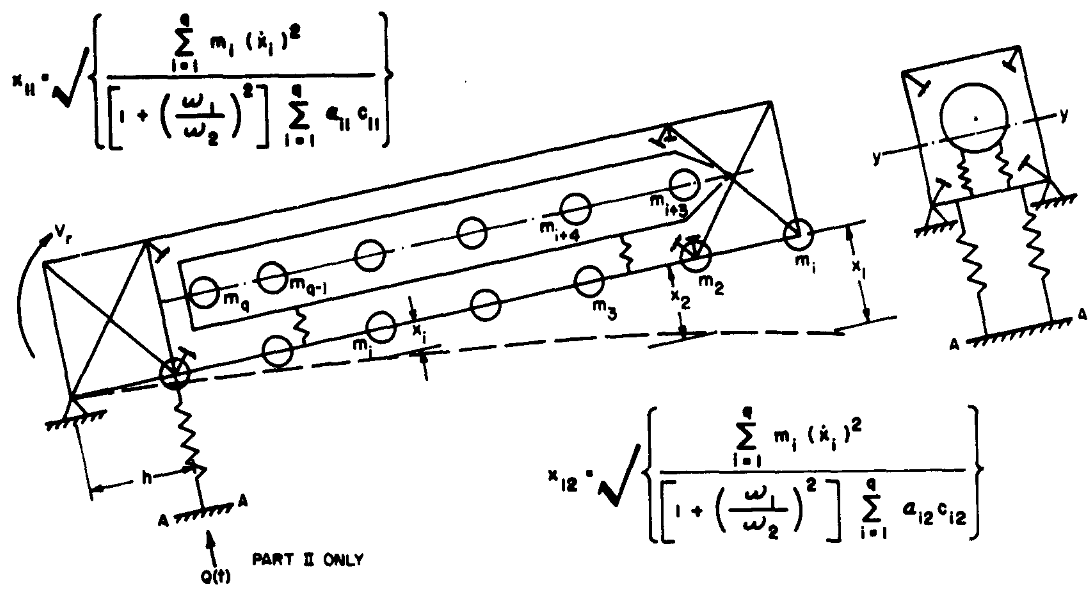

Fig. 10 - Analytical model of missile-tower system

Restrictions of the above formula were that the chord area per truss should be greater than 4 times the diagonal area and that the width-to-height ratio of the panel should be between 1/2 and 2. The flexural tower frequencies were computed from the following matric formulation which is derived in Ref. 3:

$$[AB - \lambda(HM - \alpha C)]\,Y = 0\,,$$

where

$$\lambda = \frac{m_1 \omega^2 \ell^3}{EI_1} = \text{eigenvalue},$$

$$\alpha = \frac{E}{k^1{}_{\cdot}G} \times \frac{1}{\ell^2}\;\frac{L_1}{A_1} = \text{shear parameter},$$

and where the subscripts refer to the tower station numbers. The matrices and symbols are defined in Ref. 3. The natural frequencies and mode shapes are shown in Fig. 11. The measured fundamental frequency of the actual structure was 1.61 cps.

A simple method was used for determining the torsional natural frequencies. Through the use of Eq. (1), it can be shown that the torsional spring constant between panels or between mass stations may be approximated as

$$R_i = \frac{2a^2 A_i E}{\ell_i}\,, \qquad (2)$$

where

R_i = rotational spring constant in in. lb/rad.

a = altitude of triangular tower cross section divided by 3,

and where the other symbols and restrictions are given under Eq. (1). For a square tower,

$$R_i = \frac{1.33\; a^2 A_i E}{\ell_i}\,,$$

where a = 1/2 side length. Exact solutions were compared with Eq. (2) for several panels, and Eq. (2) was found to be a reasonable approximation for design analysis purposes. The natural frequencies and mode shapes were computed from

$$(J - \omega^2 S)\Phi = 0,$$

where the matrices J, S, and Φ are defined under Eq. 13 of Ref. 4. The idealized model

[3]Cox, H. L., "Vibration of Missiles, Part II: Vibration of Missiles on Launch Stands," Aircraft Engineering, Vol. 33, No. 384, 48-55 (Feb. 1961).

[4]Cox, H. L., "A Matric Formulation of Linearly Coupled Vibration Problems," Aircraft Engineering, Vol. 30, No. 353 (July 1958).

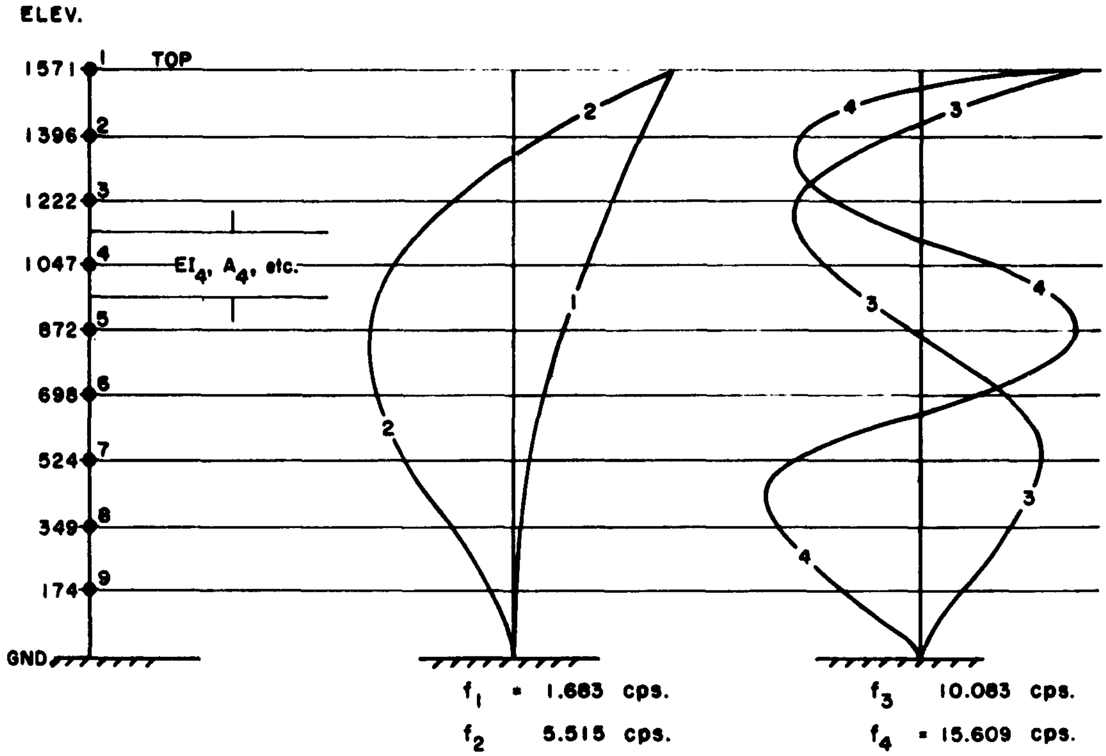

Fig. 11 - First stage umbilical tower flexural frequencies and mode shapes

and first three torsional mode shapes are shown in Fig. 12. The fundamental torsional frequency of the actual structure was measured to be 2.70 cps. Once confidence was established in the basic dynamic models, it was straightforward to compute the maximum loads induced by boom rotations and wind gusts. A modal energy distribution method, similar to the one discussed in Ref. 2, was employed.

TRANSTAINER* TESTS

In order to establish hardware design criteria for the TITAN I transtainer, both analytical analyses and experimental field tests were conducted. Analytical analyses included studies using analog and digital computers. Field tests included a shock strut orifice development test and three local road tests.

Early in the TITAN R&D program a series of shock strut orifice development tests were made to insure that the struts would provide adequate shock isolation to the missile from the expected transportation environment. The shock strut was designed to use both oil and air pressure on either side of a floating piston. Figure 13 shows cross-sectional views of the strut in the static, retracted, and extended positions. The orifice controlled the flow of oil during strut retraction and extension, which in turn determined the load factor in the transtainer and missile. Military Specification MIL-M-8090 (Mobility Requirements, Ground Support Equipment, General Specification for) was used as a design guide. According to this specification, the transtainer should comply with the requirements of Type III mobility. For example, on level paved highway, the average vehicle speed should be 50 mph. With this in mind, it was felt that the wheels should be able to pass over a series of 3-inch obstacles at 50 mph without exceeding a 3-g vertical load factor on the sprung mass. This load factor was the design load limit of the missile in a lateral direction. Figure 14 shows the test setup of the suspension system truck assembly. The weight on the wheels and strut assembly could be varied to simulate an empty or fully-loaded transtainer. Instrumentation was set up so that

*Transtainer — a combination transport vehicle and environmental protective container for the missile.

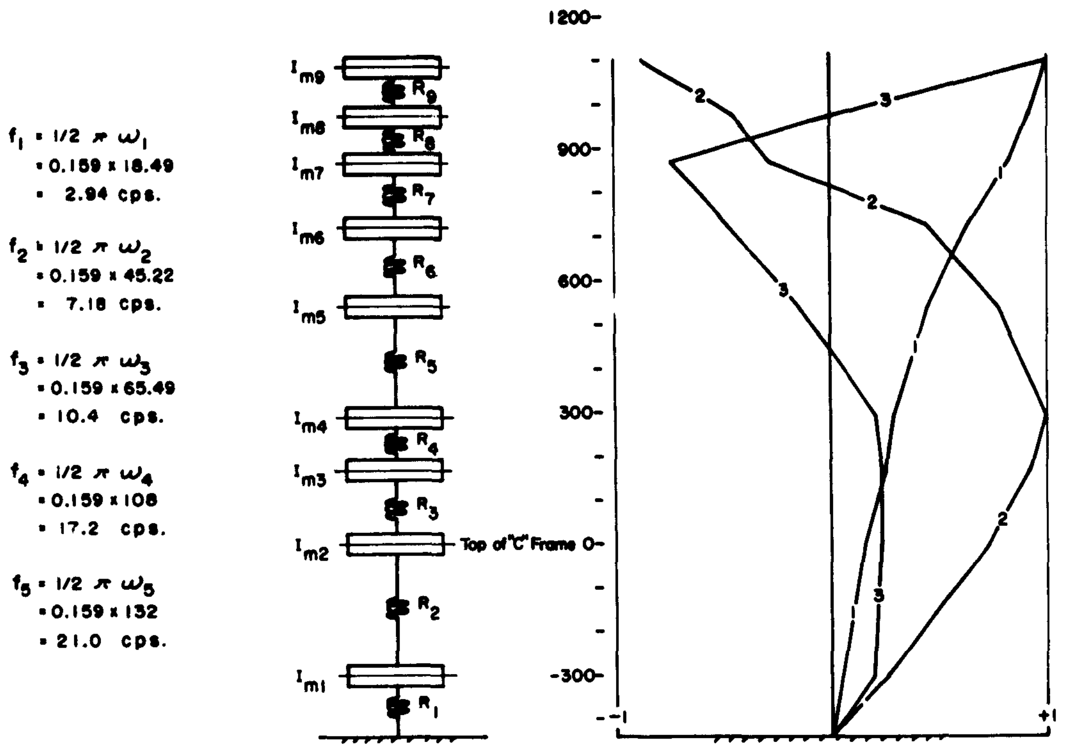

Fig. 12 - First stage umbilical tower torsional frequencies and mode shapes

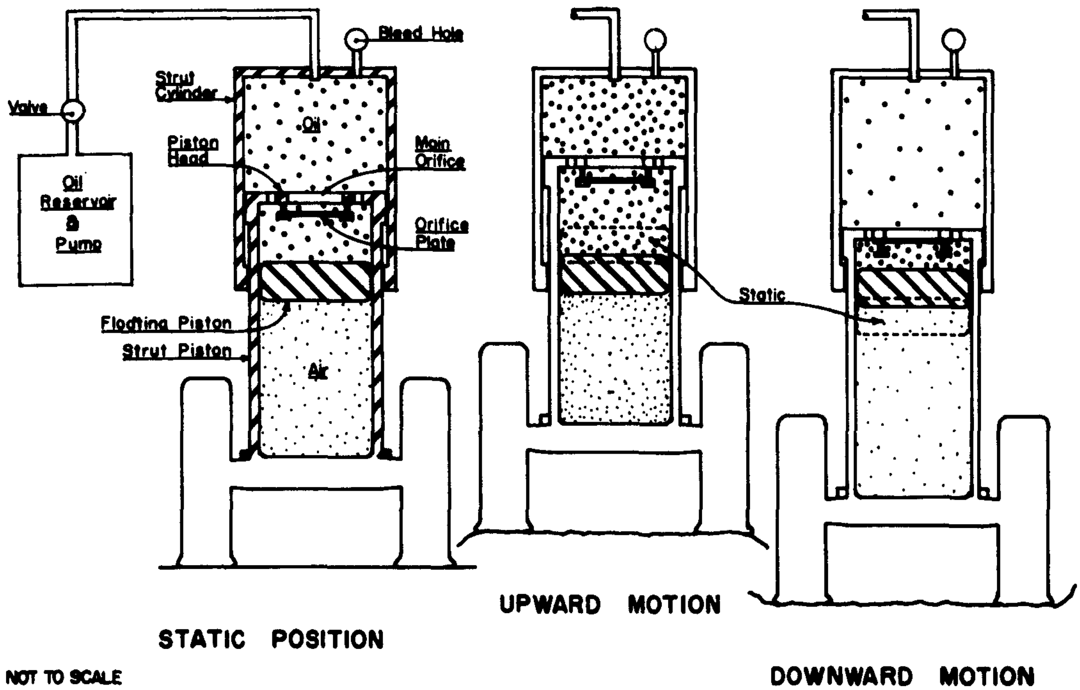

Fig. 13 - Transtainer strut operation

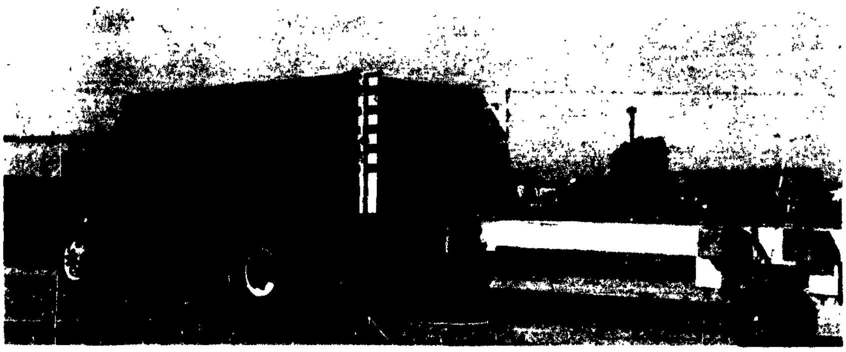

Fig. 14 - Transtainer shock strut test setup

the load factors on the sprung mass and also on the unsprung mass could be measured. Figure 15 shows the results of a typical obstacle run. The results from each set of runs determined the design changes required for each succeeding set of runs. Thus, the final orifice configuration, which gave optimum shock isolation protection to the missile, was established.

In establishing design criteria for the transtainer, another important consideration was location of the wheel assemblies. Ideally, the wheels should be located at the node points of the transtainer free-free fundamental mode shape. Such locations would minimize excitation of the transtainer fundamental bending mode from any road inputs at the wheels. With

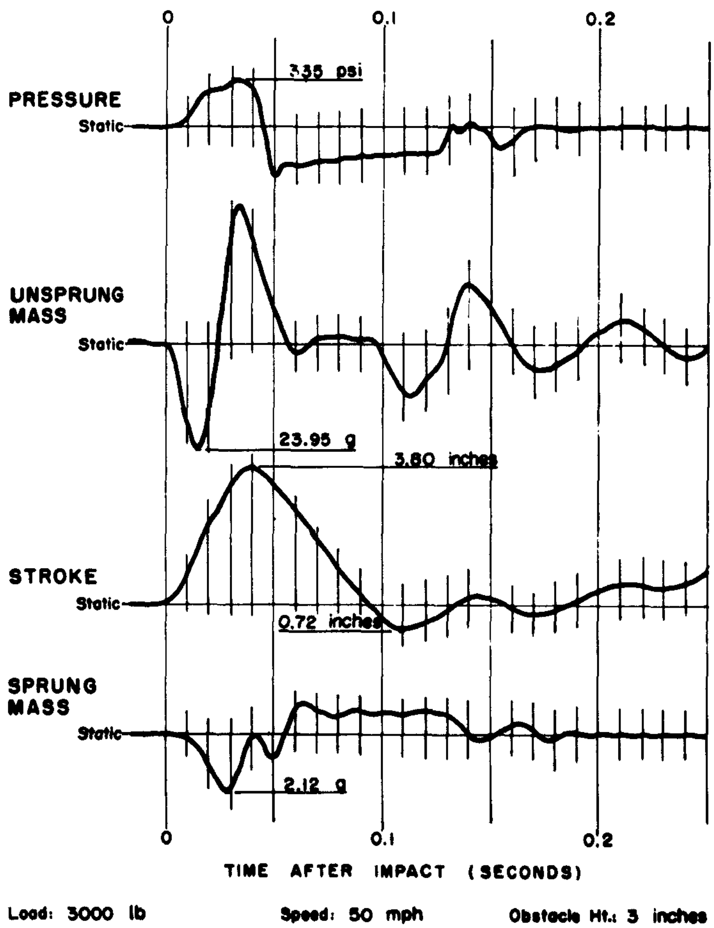

Fig. 15 - Transtainer shock strut test data

the aid of a digital computer, an eigenvalue and mode shape analysis was made on each transtainer stage in the free-free condition. Results showed that the fundamental frequency was 6.5 cps for Stage I and 24 cps for Stage II. These values were well below calculated fundamental frequencies of the empty missile stages in a free-free condition. For Stage I transtainer the node points were found to be about 5 feet from the forward end, and 4 feet from the aft end. For Stage II, the node points were about 5 feet from each end. Thus, the locations for the wheel assemblies were established; however, since the wheel wells had to be located between frame bulkheads, it was impossible to meet this design aim exactly. In the final design the wheel assemblies were located within 2 feet of the node points.

An analog study of the dynamic characteristics of the missile-transtainer system was made to help establish design criteria by predicting the response of the missile to various vibratory and step inputs at the transtainer wheels. The idealized spring-mass-damper system was the same for both stages (Fig. 16). Parameter values differed for each stage, however. The system is seen to have 6 degrees of freedom. Both the missile and transtainer were treated as rigid bodies and were allowed to deflect vertically and rotate (pitch). The unsprung mass could deflect vertically only. Masses and mass moments of inertia were readily available from a weights analysis. Spring rates, k, and damping coefficients, c, were as follows: 1 and 2 represented the rubber isolators between the missile support rings and the transtainer; 3 and 4 represented the air-oil shock struts; and 5 and 6 represented the tires. The shock strut spring rate was nonlinear because of the air spring characteristics. It can be shown that k_3 and k_4 are functions of the stroke, s, from a static position. The orifice plate in the shock strut caused higher damping, during strut extension, than compression. On the analog computer, variable damping was accomplished by switching in different damping coefficients depending on the sign of

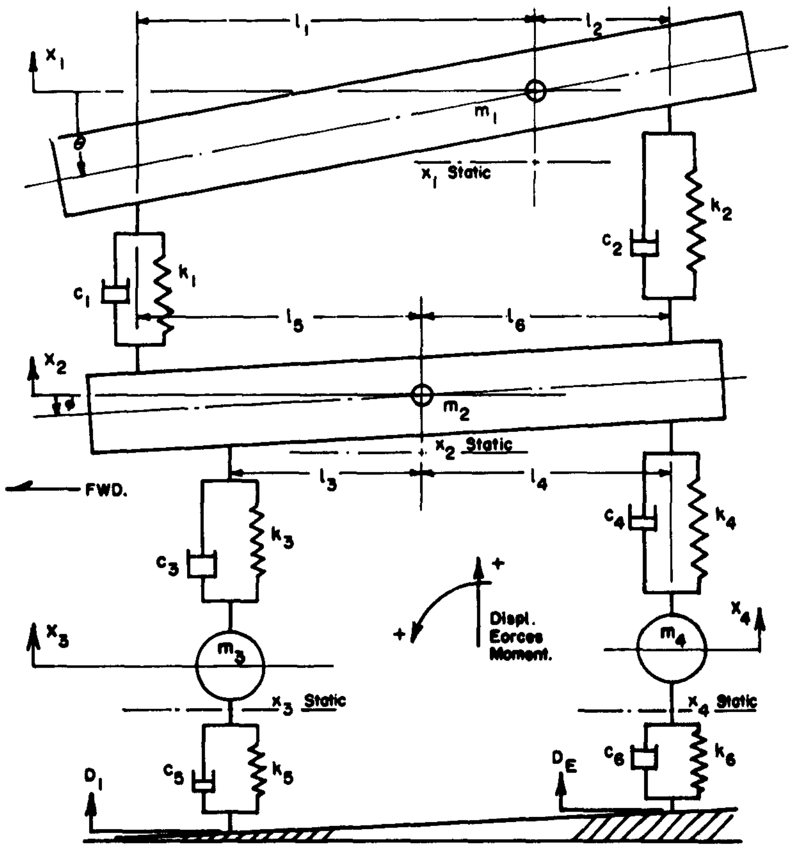

Fig. 16 - Idealized missile-transtainer system

the strut-stroke velocity. Assuming the system to deflect and rotate in the positive directions as shown in Fig. 16, the six equations of motion are:

1. $$\begin{aligned} m_1\ddot{X}_1 = &- K_1(X_1 - L_1\theta - X_2 + L_5\phi) \\ &- C_1(\dot{X}_1 - L_1\dot{\theta} - \dot{X}_2 + L_5\dot{\phi}) \\ &- K_2(X_1 + L_2\theta - X_2 - L_6\phi) \\ &- C_2(\dot{X}_1 + L_2\dot{\theta} - \dot{X}_2 - L_6\dot{\phi})\,. \end{aligned}$$

2. $$\begin{aligned} I_{m1}\ddot{\theta} = &\ K_1L_1(X_1 - L_1\theta - X_2 + L_5\phi) \\ &+ C_1L_1(\dot{X}_1 - L_1\dot{\theta} - \dot{X}_2 + L_5\dot{\phi}) \\ &- K_2L_2(X_1 + L_2\theta - X_2 - L_6\phi) \\ &- C_2L_2(\dot{X}_1 + L_2\dot{\theta} - \dot{X}_2 - L_6\dot{\phi})\,, \end{aligned}$$

3. $$\begin{aligned} m_2\ddot{X}_2 = &\ K_1(X_1 - L_1\theta - X_2 + L_5\phi) \\ &+ C_1(\dot{X}_1 - L_1\dot{\theta} - \dot{X}_2 + L_5\dot{\phi}) \\ &+ K_2(X_1 + L_2\theta - X_2 - L_6\phi) \\ &+ C_2(\dot{X}_1 + L_2\dot{\theta} - \dot{X}_2 - L_6\dot{\phi}) \\ &- K_3(X_2 - L_3\phi - X_3) \\ &- C_3(\dot{X}_2 - L_3\dot{\phi} - \dot{X}_3) \\ &- K_4(X_2 + L_4\phi - X_4) \\ &- C_4(\dot{X}_2 + L_4\dot{\phi} - \dot{X}_4) \end{aligned}$$

4. $$\begin{aligned} I_{m2}\ddot{\phi} = &- K_1L_5(X_1 - L_1\theta - X_2 + L_5\phi) \\ &- C_1L_5(\dot{X}_1 - L_1\dot{\theta} - \dot{X}_2 + L_5\dot{\phi}) \\ &+ K_2L_6(X_1 + L_2\theta - X_2 - L_6\phi) \\ &+ C_2L_6(\dot{X}_1 + L_2\dot{\theta} - \dot{X}_2 - L_6\dot{\phi}) \\ &+ K_3L_3(X_2 - L_3\phi - X_3) \\ &+ C_3L_3(\dot{X}_2 - L_3\dot{\phi} - \dot{X}_3) \\ &- K_4L_4(X_2 + L_4\phi - X_4) \\ &- C_4L_4(\dot{X}_2 + L_4\dot{\phi} - \dot{X}_4)\,, \end{aligned}$$

5. $$\begin{aligned} m_3\ddot{X}_3 = &\ K_3(X_2 - L_3\phi - X_3) + C_3(\dot{X}_2 - L_3\dot{\phi} - \dot{X}_3) \\ &- K_5(X_3 - D_1) - C_5(\dot{X}_3 - \dot{D}_1),\ \text{and} \end{aligned}$$

6. $$\begin{aligned} m_4\ddot{X}_4 = &\ K_4(X_2 + L_4\phi - X_4) + C_4(\dot{X}_2 + L_4\dot{\phi} - \dot{X}_4) \\ &- K_6(X_4 - D_2) - C_6(\dot{X}_4 - \dot{D}_2)\,. \end{aligned}$$

The symbols used in these equations are defined in Fig. 16. When the applicable masses, spring rates, damping coefficients, and geometry were substituted into these equations, the problem was sent to the analog computer for analysis. In determining the dynamic response of the missile, three types of inputs at the wheels were considered — sinusoidal, obstacle, and ramp. These inputs were either in-phase or out-of-phase at the front and rear wheels. The response of the Stage II missile-transtainer system was found to be quite similar to that of the Stage I system, with the exception that the former was characterized by pitching amplitudes which were large in relation to the latter. This condition was to be expected since the spacing of the Stage II support rings was closer than that of the Stage I. When the wheels rolled over a 3-inch high obstacle, in the analog study of both systems, load factors of up to 3.5 g were recorded on the missile at speeds of 35-50 mph. It was felt that accelerations found from this study would be somewhat higher than those from actual experiments. This fact was borne out in later road testing. It was concluded that strut damping factors of 5 percent of critical for retraction and 15 percent of critical for extension should give adequate damping to prevent excessive amplitudes of the missile and to keep missile accelerations below the maximum allowable limits.

The objectives of road testing the missile-transtainer systems were first, to demonstrate type-III mobility (Military Specification MIL-M-8090) over paved highways at speeds up to 60 mph and over gravel roads at speeds up to 25 mph; and second, to determine isolation characteristics of the transtainer suspension system. The prime mover for towing the transtainer was an Army 6 × 6 truck. Tri-axial accelerometers and impact meters were mounted at critical points on the system. The accelerometers, wired to an oscillograph, recorded the time history of any road input desired while the impact meters provided a continuous record of shock levels in 0.5-g increments. Each transtainer was road tested over approximately 500 miles of paved highway and gravel road. Results showed that the missile acceleration limits were not exceeded at any time. However, accelerations of 4 g were common on the Stage I engine bell. It was found that the engine bell lateral frequency was higher than anticipated. Both the missile rigid body and the engine lateral frequencies seemed to lie in the same range of 12 to 15 cps. Exact frequencies could not be isolated because of the continuous random input at the wheels. As a result of this test, the criteria were changed to include more rigid locking links between the engine bells and the missile.

The size 7.50 × 10 tires on the Stage I transtainer tended to overheat during most of the road test, because of loading beyond their rated capacity. As a result, the Stage I transtainer criteria were changed to include a larger tire, size 9.00 × 10. The small 10-inch diameter of the tires was necessary in order to clear some highway underpasses. In general the dynamic load levels recorded during the road tests were lower than the levels predicted from analytical calculations partly because actual weights were from 7- to 25-percent higher than the weights used in the analytical analyses. Theoretical and experimental results from the TITAN I R&D program were used to help establish the design criteria for the TITAN II transtainers. Compared to TITAN I, Stage I of the TITAN II missile was slightly longer and heavier; the diameter of Stage II was changed from 8 to 10 feet, or the same as Stage I. This growth capability was originally built into the Stage II transtainer so that the only major modification was a repositioning of the four missile ring support points. As a result of extensive theoretical calculations and road testing of the TITAN I transtainers, a substantial reduction in time and expense was realized on the TITAN II transtainer development.

MISSILE SUSPENSION SYSTEM

The missile suspension system for TITAN II was designed to shock-isolate the missile in the silo from nuclear blast effects. Figure 17 shows the missile mounted on the suspension system in the silo. The basic criteria for the TITAN II missile suspension system resulted from experience gained on the TITAN I program. Some of the more important criteria items are as follows:

"Missile Alignment Requirements

"After sustaining simulated or actual nuclear blast effects, capability must exist for locking the missile thrust mount into the position for firing, and after the missile is locked into the firing position the support plane encompassing the missile attachment points shall

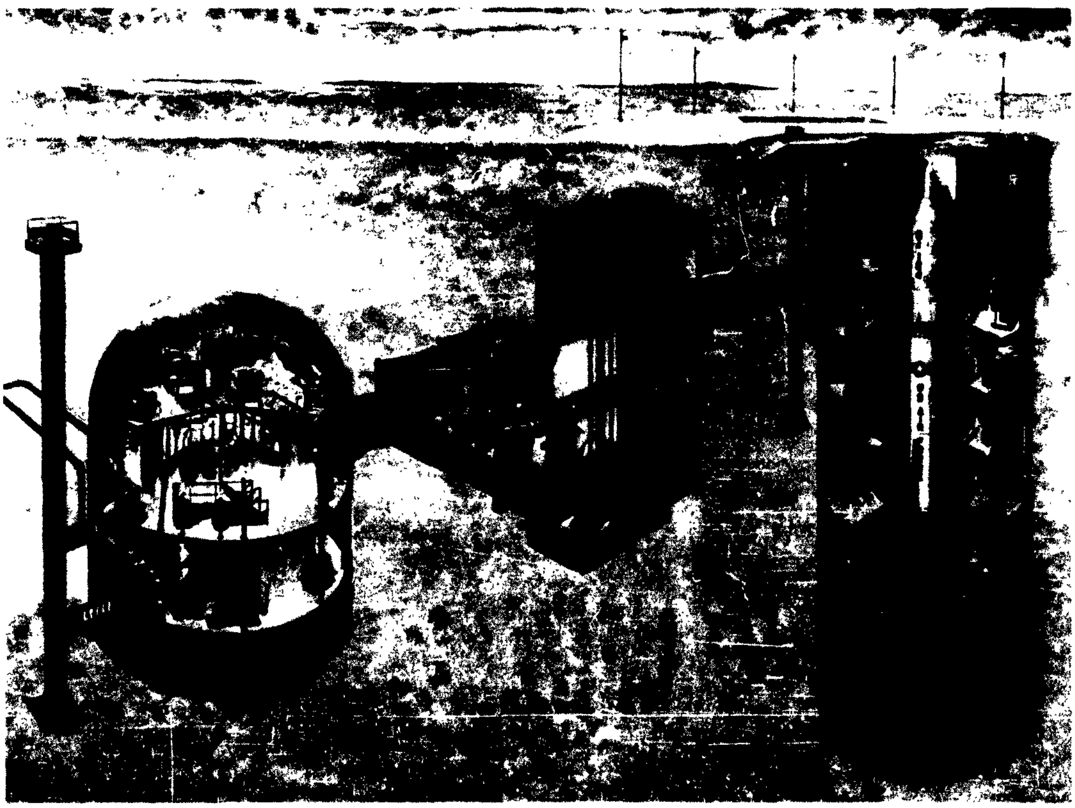

Fig. 17 - TITAN II operational complex

not deviate from the true horizontal by more than 0.1 degrees. Prior to receiving nuclear blast effects or in the normal hard readiness condition, the support plane encompassing the missile attachment points shall not deviate from the true horizontal by more than 0.05 degrees.

"Azimuth - The missile supporting system shall position the missile along a given azimuth line to within a tolerance of ± 0.25 degrees, prior to and after nuclear blast.

"Vertical - The missile supporting system shall maintain the plane of the missile supports to within a tolerance of ± 0.25 degrees from the horizontal under the most severe conditions of loading resulting from both static and dynamic loads.

"Adjustment - No support points shall deviate from a true horizontal plane by more than ± 0.1 inch when the adjustable support mounts are in the neutral positions. The missile support system shall provide adjustment means to place each of the four support points within the support plane to a tolerance not to exceed ± 0.01 inch. The support arm attachments shall be capable of ± 1/2 inch adjustment in the vertical direction, ± 1/2 inch tangentially, and ± 1/2 inch radially.

"Structural Stiffness Requirements

"Hard Position - The thrust mount support structure and shock isolation system shall be designed to develop the following total influence coefficients at the attachment points:

$a_{jj} = 32 \times 10^{-6} \frac{\text{in.}}{\text{lb.}}$ (vertical)

$a_{jj} = 3.67 \times 10^{-9} \frac{\text{rad.}}{\text{in.lb.}}$ (pitch and yaw rotational)

$a_{jj} = 1.04 \times 10^{-3} \frac{\text{in.}}{\text{lb.}}$ (horizontal)

$a_{jj} = 57 \times 10^{-9} \frac{\text{rad.}}{\text{in.lb.}}$ (torsional)

"The maximum spring constant difference that shall exist between any two vertical spring assemblages shall be two percent.

"Firing or Launch Position -

$a_{jj} = 1.3 \times 10^{-6} \frac{\text{in.}}{\text{lb.}}$ (horizontal)

$a_{jj} = 0.86 \times 10^{-6} \frac{\text{in.}}{\text{lb.}}$ (vertical)

$a_{jj} = 0.161 \times 10^{-9} \frac{\text{rad.}}{\text{in.lb.}}$ (pitch and yaw rotational)

$a_{jj} = 20 \times 10^{-9} \frac{\text{rad.}}{\text{in.lb.}}$ (torsional)

"Locking and Unlocking Mechanism Requirements

"When nuclear blast induced motion has been attenuated to ± 0.1 inches in the horizontal and vertical directions, locking up of the thrust mount structure to the silo walls may begin. The total lock up time shall be less than one tenth of a second. After a missile has been launched and it is necessary to unlock the thrust mount, during the unlocking, the missile support plane at the missile attachment points shall not deviate from a true horizontal plane by more than ± one-quarter degree. The one-quarter degree deviation applies to a smooth unlocking procedure; if the unlocking procedure is pulsating, the maximum deviation of the missile support plane in the launch position shall be less than 0.1 degree.

"GOE Locking Mechanism Requirements - When the locking pins have securely been engaged, an electrical signal will be sent to the facilities portion of the console which will indicate that the automatic lock up for launching has been completed, and status of the lock up mechanism will be shown at any time.

"Manual Back-Up Locking Mechanism - In the event the automatic lock up mechanism fails, a manual override of the locking mechanism system will be provided.

"Shock Mount Damping Characteristics

"Vertical - A total of four coulomb dampers which develop a kinetic force within each damper of approximately three hundred pounds shall be used. Provision shall be made to provide a slip joint which can reduce the damping force to zero if desired.

"Horizontal - Two coulomb dampers will resist horizontal motion in either the pitch or yaw directions. Each of these two dampers shall develop a kinetic force of approximately fifty pounds. Provision shall be made to provide a slip joint which can reduce the damping force to zero if desired.

"Torsional - The horizontal dampers will automatically establish the torsional damping characteristics.

"Adjustability - The horizontal dampers shall provide adjustment which will enable each damper to provide a kinetic damping force of thirty pounds to two hundred pounds. Each vertical damper shall have adjustment provisions which will provide for kinetic damping forces between two hundred fifty pounds and one thousand pounds. A lengthening and shortening adjustment of at least ± 2 inches shall be provided for all horizontal and vertical dampers.

"Environment - The thrust mount and shock isolation system shall operate satisfactorily as defined in AFBM Exhibit 60-12, Environmental Specification for the Ground System of WS-107A-2, Titan II, dated 22 April 1960.

"Hold Down Requirements - The operational missile will not require special hold-down devices such as bolts or clamps to prevent missile damage during the response to nuclear blast effects.

"Operational - The criteria expressed herein will govern for operational systems.

"Interface Data

"Spring Constants or Stiffness Requirements at Silo Wall Interfaces - The vertical spring constant developed at each of the four vertical spring attachment points shall be equal to or greater than 0.8×10^6 pounds per inch. The vertical spring constant at each of the four vertical lock up attachment points must be equal to or greater than 0.9×10^6 pounds per inch. The spring constant requirements at each of the horizontal attachment points for a load directed along the line of the lock up strut shall be equal to or greater than 1.8×10^6 pounds per inch.

"Umbilical Misalignment Contributions - Total umbilical forces shall not cause the missile to be misaligned from the true vertical by more than 0.03 degree."

Discussion

The maximum design dynamic loads were for malfunction firings, and much of the data discussed in the first section of this paper was used in establishing the final design loads. The criteria were basically a compromise to account for dynamic loads imposed by blast, damping decay time so that lockup could occur, stability, and ability to return to the initial vertical position after sustaining blast. Since the missile-damper-attachment-points, at the silo walls, moved with rather high velocity as a result of nuclear blasts, it was decided to use coulomb dampers in order to prevent partial damper strut lockup that could occur with viscous damping.

A breadboard damper was built and tested under sinusoidal and impact loads in order to de-bug the design before final engineering release. Also, a complete system was assembled and a tower, which simulated a missile, was installed. Damping decay times, ability to lockup, spring constant tolerances, and natural frequencies were checked. As a result of these tests, the final damping values and lockup signal requirements were selected and incorporated into the final engineering procurement specifications.

The dynamic model for predicting the response of the missile and suspension system to ground motions caused by nuclear blasts is shown in Fig. 18. Since the missile flexural frequencies were high, compared to rigid body frequencies, only rigid body motion was considered. This motion was defined by x_1, x_2, and θ_3 as shown. The equations of motion for the system were solved by the technique discussed in Ref. 5. The typical response for this type of system is shown in Fig. 19. Note that the horizontal and rotational motion is much slower than the vertical motion. By the time that maximum horizontal motion is attained, the vertical motion has been damped considerably.

SILO LAUNCH

A full scale silo was constructed for the purpose of determining, from captive and flight test, basic data regarding air entrainment, acoustic and vibration levels, engine start transient overpressure, thermal effects, and exhaust-duct-liner requirements. The silo configuration is shown in Fig. 20. Careful analysis of the data obtained from the captive firing and launch established the basic design criteria for the operational TITAN II silos (Fig. 17).

Some of the basic data, obtained along with extrapolations to the TITAN II concept, are given in Table 2.

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[5]Chan, S. P., Cox, H. L., and Benfield, W. A., "Transient Analysis of Forced Vibrations of Complex Structural Mechanical Systems," J. Roy. Aeronaut. Soc., 66:457-460 (July 1962).

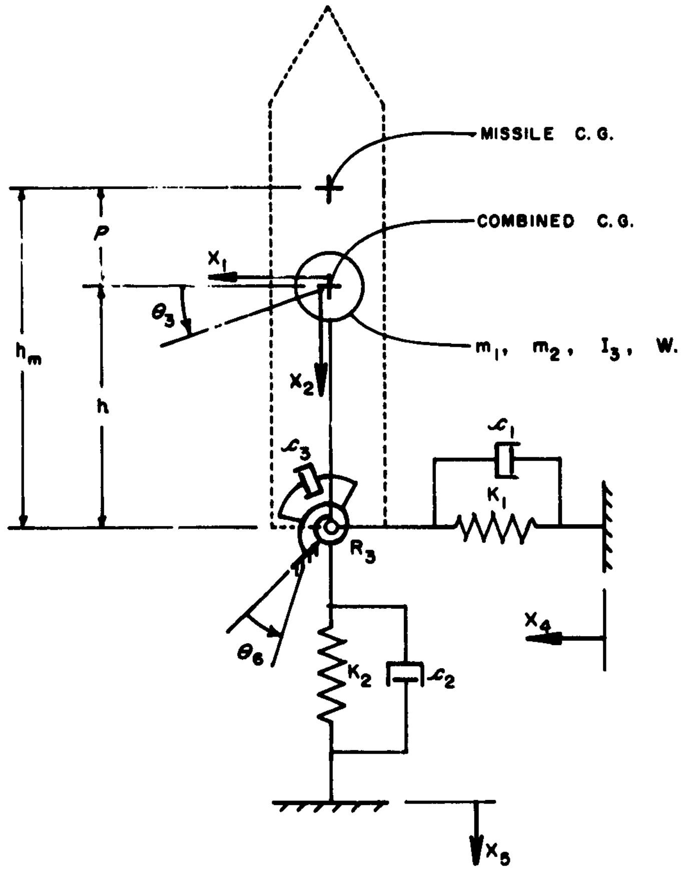

Fig. 18 - Dynamic model, hard condition

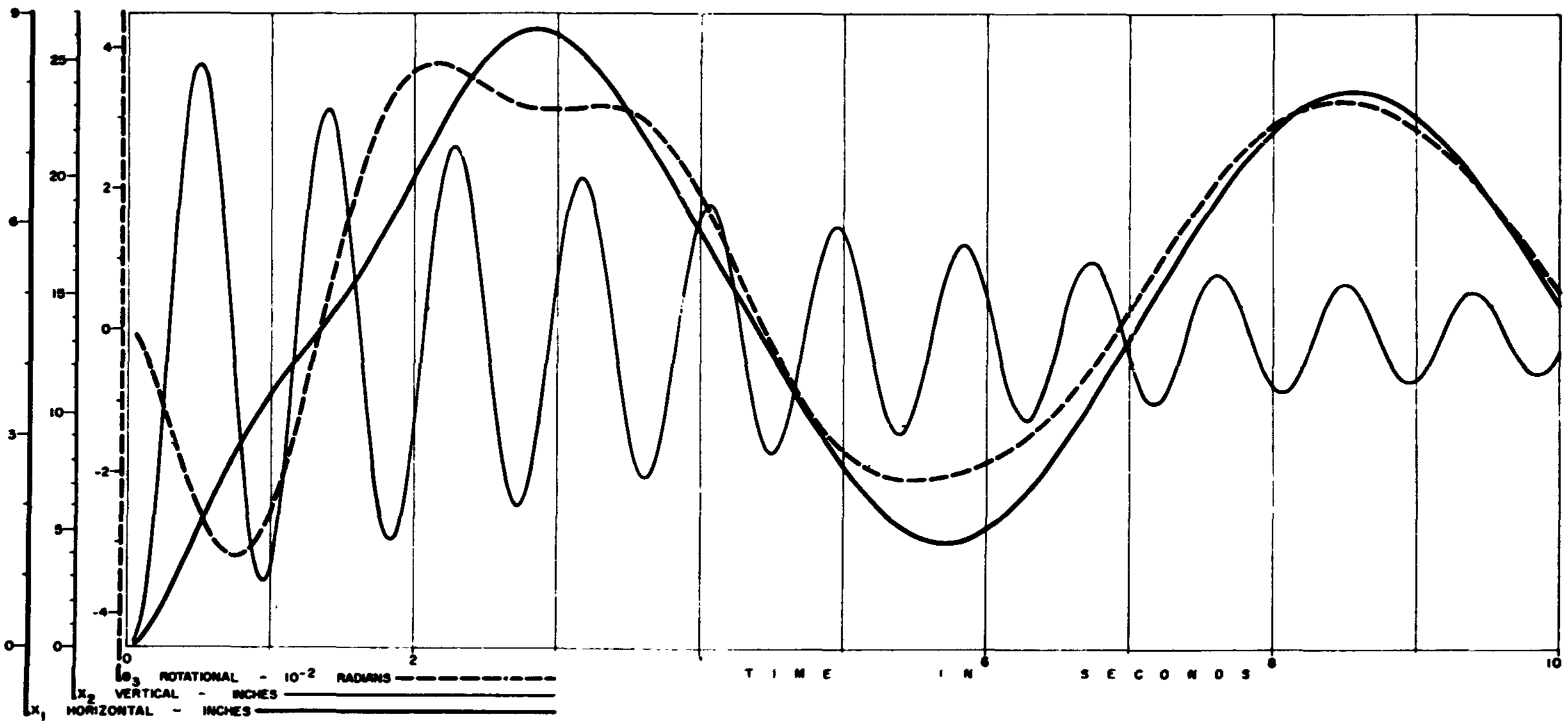

Fig. 19 - Typical response to ground shock (rigid body motion)

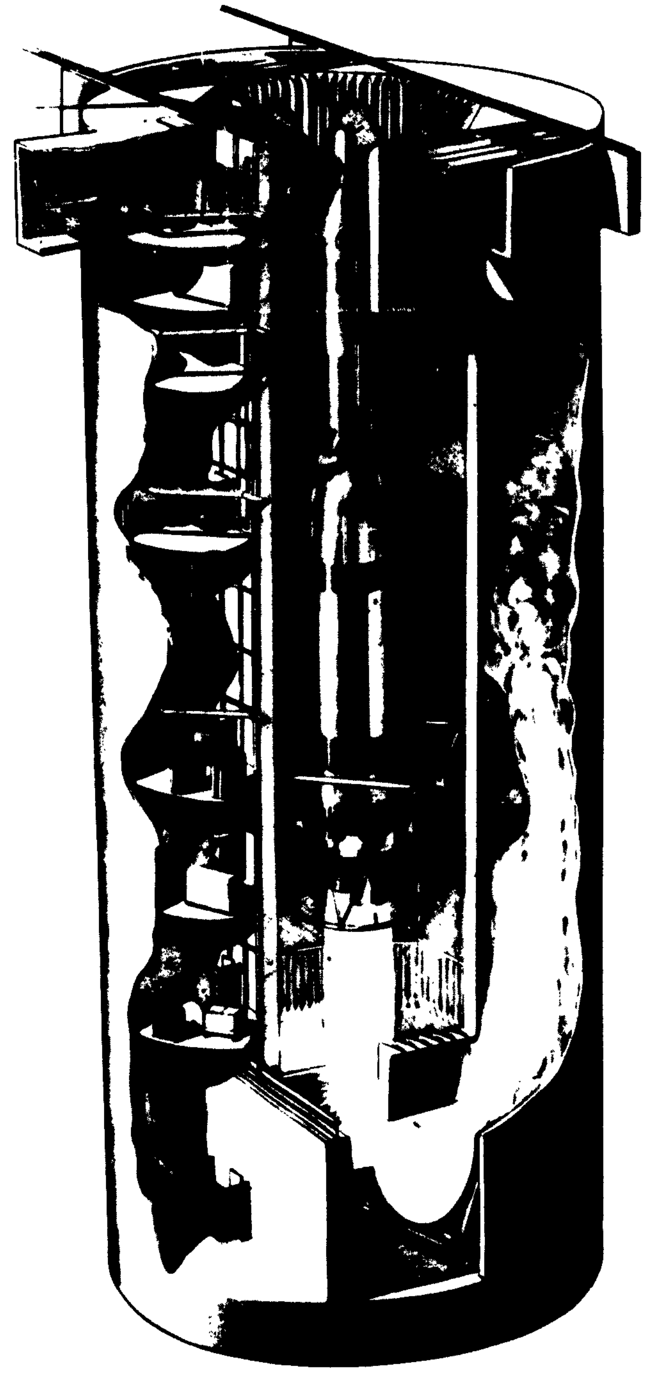

Fig. 20 - TITAN I silo launch configuration

TABLE 2
Silo Launch Test Facility Data and TITAN II Extrapolations

Item		Silo Launch Test Facility (3 feet of launch-duct-liner)	TITAN II Operational Silo Predictions (1 foot of launch-duct-liner plus exhaust-duct-liner)	Remarks
Total Acoustic Levels on Missile	Tail Skirt Transition Nose Cone	156 db 145 db 158 db	158 db 149 db 150 db	O.K.
Above Ground Total Free Field Acoustic Levels (feet)	0 10 20 30	146 db 152 db 150 db 147 db	146 db 144 db 142 db 140 db	TITAN II to have exhaust duct liner.
a = Air Entrainment		3.1	1.9	Somewhat low for TITAN II
Launch Duct Overpressure		<4 psi	<3 psi	Water injection into exhaust for TITAN II
Exhaust Duct Overpressure		4.5 psi	<4 psi	No problems
Vibration Levels within Missile		Most equipment measured levels below test failure levels	Some special mounting of a few components	
Missile Structural Strength		No structural failures of any type	No structural failures of any type	No problems
Exhaust Duct Temperatures		3000°-4000°F	<2000°F (water injection)	No problems
Fundamental Missile Lateral Natural Frequency		0.701 cps	0.370 cps	O.K.

CONCLUSIONS

Many examples have been presented to show how the use of measured data, obtained from component and full scale tests, helped establish or revise design criteria for primary ground support structures in the TITAN R&D program. In most cases, final design criteria for operational hardware could not be established until comprehensive tests were made and the test data analyzed early in the program. It has been shown that malfunction engine firings can reveal valuable data for rewriting design criteria or establishing criteria for follow-on programs. For the missile silo launch concept, a full scale weapons system was built to prove the feasibility of underground missile launchings. The use of measured data from extensive test programs to establish design criteria, has reduced development time, increased reliability, and reduced the total cost of the R&D program.

DISCUSSION

Mr. Christensen (Aerojet): I would just like to commend these gentlemen on the extraordinarily good presentation, such a cooperative effort there of the visual aids and the good timing, extraordinary.

Mr. Nankey (GE): I would like to know how you excited your full scale dynamic model?

Dr. Cox: Of the missile itself, we excited only the fundamental mode, primarily as I pointed out, to check the mathematical model, the mass distribution, springs and so on. This was done purely by hand. We walked up and pushed the missile back and forth. The missile fully loaded is quite heavy; however, by pushing at the natural frequency and getting the rhythm right, we were actually able to excite the missile by hand. The erector tower which weighs 260,000 pounds, we excited also in the fundamental mode, simply by standing on top and shifting our weight back and forth, right at resonance, this requires quite a bit of timing. I suppose people who are good twisters might be more adept at this than some of us engineers.

Mr. Floury (Korfund): Could you tell us why the coulomb damper was chosen for the TITAN II suspension system?

Dr. Cox: We had the problem of making the missile motion decay out to be compatible with the overall weapons system launch time requirements and, of course, damping was required for this purpose. Now the velocity of the wall (or the shock spectra input) is very high, and a typical viscous damper may tend to lockup. One would have to build complicated relief valves or bypass valves, and so on, because of the very high velocity. The damping, of course, being roughly proportional to the velocity. So it was decided that a coulomb damper would do the trick a lot better, since it is essentially independent of the velocity.

* * *

CALCULATION AND SIMULATION OF THE NOISE ENVIRONMENTS OF A GUIDED MISSILE FOR TESTING OF COMPONENTS*

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This paper describes the acoustic testing of a selected missile subsystem. The methods used for calculating the external noise fields, originating from the booster engine and the turbulent boundary layer of a supersonic missile, are discussed. The overall noise level and the spectrum of the booster noise were calculated from similarity laws. The calculation of the same quantities of the boundary layer noise was based on the local properties of the induced flow. A large reverberant chamber and a spectral synthesizer were used for simulating the calculated noise environments. After measuring the noise reduction through the structure of the missile, suspended inside the reverberant chamber, the internal noise spectrum was calculated and synthesized for functional and fatigue tests of an arm safe device and its components. The behavior during operation was recorded and analyzed.

INTRODUCTION

The objective of this program was to determine the reliability of certain missile components of a selected missile subsystem when subjected to a noise environment. During the course of the program, it was determined that, with one exception, all subsystem components performed satisfactorily. The three major phases of the program were as follows: first, a calculation of the most severe noise fields; second, a selection of a simulation technique to simulate these noise fields at conditions close to that predicted in Phase I; and third, the test phase and its evaluation.

The sequence of this program is typical of all tasks requiring knowledge of an operational environment and its effect on certain components during the early design stage of a missile, if experimental values are not yet available.

An estimate of the overall noise levels during a complete flight of the missile indicated that two noise sources would contribute to this environment: booster engine noise prevailing only during subsonic flight, and noise from the turbulent boundary layer, during the entire flight in the atmosphere.

These two noise sources are closely related to certain flight parameters such as altitude, velocity, and free-stream dynamic pressure; these are shown in Fig. 1 for the ascent phase of the missile. The time function of these parameters is typical for a guided missile, operating at low altitude. Of particular interest

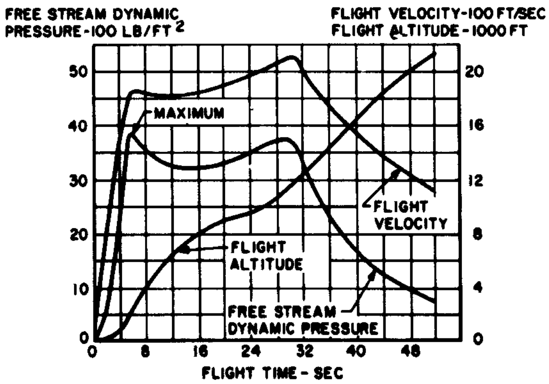


Fig. 1 - Flight parameters of a supersonic missile during ascent

*This program was conducted under the auspices of Picatinny Arsenal, Dover, N. J., under Contract DA-30-069-501-ORD-3186.

is the dynamic pressure which increases sharply after boost; it has a broad maximum for approximately 30 seconds, and decreases slowly with increasing altitude.

QUANTITIES OF AN EXTERNAL NOISE FIELD

It is common practice to describe an external noise field around a vehicle by three quantities:

- A time-averaged overall noise level,

- A time-averaged spectrum or octave band level, and

- A spatial correlation between the instantaneous pressures.

The following paragraphs discuss the methods used to predict the overall noise levels and the spectra of the two main noise sources. No attempt was made to predict the spatial pressure correlation, i.e., the phase correlation between the instantaneous pressures acting over the outer wall of the compartment, because the results of spatial pressure correlation of rocket engines for points located upstream of the nozzle exit are insufficient and do not allow application of any similarity consideration. Furthermore, the components to be tested are located inside the compartment which has an irregular surface of high reflection that causes the build-up of a reverberant field. Certain standing waves at discrete frequencies might also exist. It can be assumed that even in the case of an external noise field with a well defined spatial correlation, the resulting internal noise field is diffuse and its spatial pressure correlation completely random. This assumption justifies the decision to restrict this study to the first two quantities.

Booster Engine Noise

The external near noise field around the missile compartment, which originated from the booster engine, was determined by extrapolation of near noise field results obtained from a similar rocket engine.

Figure 2 shows the overall noise level as a function of the dimensionless axial distance X/D for a solid propellant rocket engine of 7000 pounds thrust measured close to the main axis at a distance of $y/D = 3$ [1].

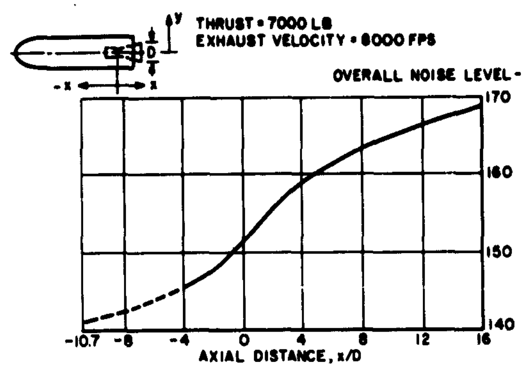


Fig. 2 - Near noise of a rocket engine, measured at a radial distance $y/D = 3$

It can be shown by a similarity consideration (Appendix A) that the near field noise-distance function holds for rocket engines of other thrust values at the same exhaust velocity, if the acoustical power coefficient is assumed to be constant. The thrust of the rocket engine was 14,000 pounds, and from the diagram, an overall noise level of 141 db at the compartment station was obtained for the missile at rest and at sea level.

This noise level will be influenced by the variation of the characteristic impedance of the ambient air and motion of the noise source during flight. During the subsonic flight phase, the decrease of the characteristic impedance was only 1 percent and was neglected. The effect of motion of the noise source on points upstream of the nozzle exit has been studied theoretically in several papers [2, 3, and 4]. Different corrections to the noise level were derived, depending on the type of elementary noise sources and the orientation of their axes relative to the direction of motion. As a result, either an increase or a decrease of the noise level has been predicted, with values approaching 30 db near Mach number one. Recent flight measurements, however, did not confirm the predicted large variation with Mach number. Only a slight decrease was found in front of jet engines flying at a high subsonic speed [5]. Consequently, the effect of motion was not taken into account.

The spectrum of the rocket engine can be determined from the spectrum of the reference engine, if one assumes the same Strouhal

number S* for both spectra in the near noise region. The resulting spectrum is presented in Fig. 3. It shows a broad maximum between the center frequencies, 425 and 850 cps.

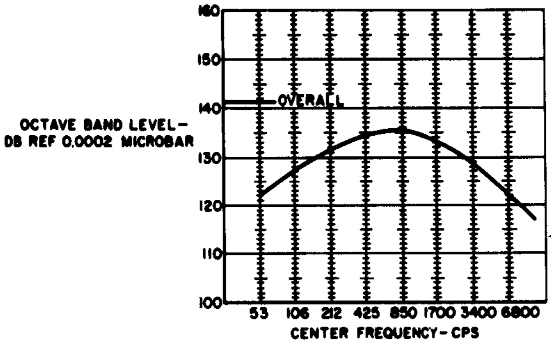


Fig. 3 - Spectrum of booster engine noise

Boundary Layer Noise

A discussion of the calculation of the second source of noise, originating from the boundary layer, follows.

The relative motion between the fluid particles of a gaseous medium such as air and the solid boundary of a missile will produce the boundary layer as shown in Fig. 4. This figure illustrates an induced flow field of a supersonic missile flying at a low altitude. The region behind the shock may be divided into the viscid and the inviscid flow regimes. The boundary layer is the viscid portion of the flow near the solid surface, in which the viscous effects of the fluid are predominant. The laminar portion of the boundary layer is characterized by a nonmixing flow, in superposed layers, where the fluid properties, as well as the particle velocity at every point, are constant with time. A transition from laminar to a turbulent flow will take place when the inertia forces of the fluid particles, within the boundary layer, become much larger than the viscous forces. The point of transition may be predicted from the nondimensional Reynold's number, defined as the product of the fluid density, fluid velocity, and a characteristic length divided by the fluids viscosity. When the local Reynold's number exceeds some critical value along the body surface, the boundary layer becomes turbulent. A turbulent boundary layer is marked by rapid fluctuations in the fluid properties which, because of their random motion, become sources of noise generation. The superimposed effect of all sources, termed the boundary layer noise, can become a major contributor to the missile's total noise environment.

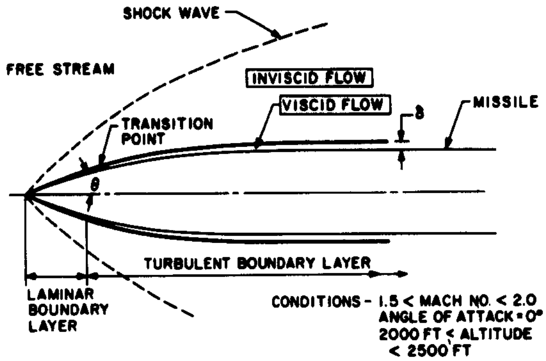


Fig. 4 - Induced flow around a supersonic missile

Overall Sound Pressure Level (SPL_{ov}) — The calculation of the overall sound pressure level was based on experimental information of the fluctuating pressure disturbances within the boundary layer. Their root-mean-square value ($\bar{p}$) has been found to be directly proportional to the local value of the dynamic pressure (q) at the boundary layer edge. In the case of a missile, the local dynamic pressure depends upon the missile's altitude, attitude, velocity, and configuration. For a subsonic missile, at zero angle of attack, the local dynamic pressure may be assumed equal to the free-stream dynamic pressure, irrespective of configuration. For a supersonic missile, at all angles of attack, the local dynamic pressure is influenced by the shock wave as shown in Fig. 5. A sudden increase in the dynamic pressure across the shock is followed by a maximum at about one-half-diameter downstream and a minimum near the shoulder. The region of maximum local dynamic pressure may well be laminar. Transition will probably occur between this region and the shoulder.

The equipment compartment of the missile is located aft of the shoulder in a region where the local dynamic pressure is nearly constant. This fact, together with the variation in the ratio $\bar{p}/q$ (from 4.5 to 10×10^{-3}), found by different authors [6, 7, and 8] would justify the use of

*The dimensionless Strouhal number S, for engine noise, is defined by the ratio $S = fD/v$, where f = peak frequency of the spectrum, D = nozzle exit diameter, v = exhaust velocity.

the free-stream dynamic pressure without appreciable reduction of accuracy. However, it was decided to determine the local dynamic pressure by a method that could be employed in the future when more accurate values of $\bar{p}/q$ will have been established. Appendix B shows how local dynamic pressure can be calculated from the total energy, entropy, and static pressure of the fluid. A previous paper [9] by one of the authors, also used the local dynamic pressure in determining the boundary layer noise of a blunt nose re-entry vehicle.

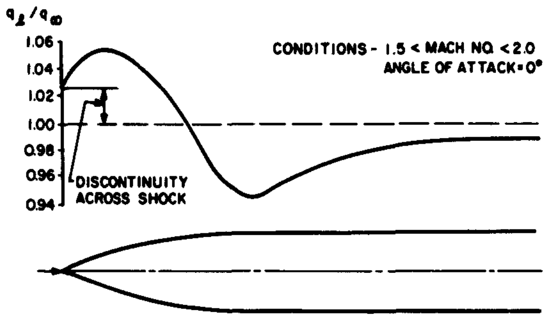


Fig. 5 - Ratio of local-dynamic-pressure to free-stream-dynamic-pressure on the surface of a supersonic missile

The local dynamic pressure varies with time as a function of the flight velocity and altitude. At zero angle of attack, the local and free-stream values of the pressure occur simultaneously. Consequently, the maximum boundary layer noise will coincide with the condition of maximum local dynamic pressure. It was decided to simulate this maximum value rather than its variation with time. Multiplication of the maximum local dynamic pressure with an average value of $\bar{p}/q$ (6.8×10^{-3}) results in a maximum overall noise level of 155.4 db around the compartment.

The spectrum of the boundary layer noise was computed from spectra obtained during the Little Joe 2 flight tests. The measured internal noise spectrum [10] and the noise reduction recorded during other flights of the same vehicle [11] permitted calculation of the external boundary layer noise spectrum, shown in Fig. 6. It is made nondimensional by multiplication with the ratio ∂_c/V (∂_c = compressible boundary layer thickness, V = flow velocity at boundary layer edge), calculated at the location corresponding to the point where the internal noise was recorded (Appendix C). It was assumed that this nondimensional spectrum is constant for different configurations over a range of subsonic and supersonic flow velocities as long as the flow conditions are similar and no adverse pressure gradients occur. Multiplication of the nondimensionalized spectrum by the ratio V/∂_c, calculated for the missile, resulted in the boundary layer noise spectrum (Fig. 7). It peaks between the center frequencies, 850 and 1700 cps.

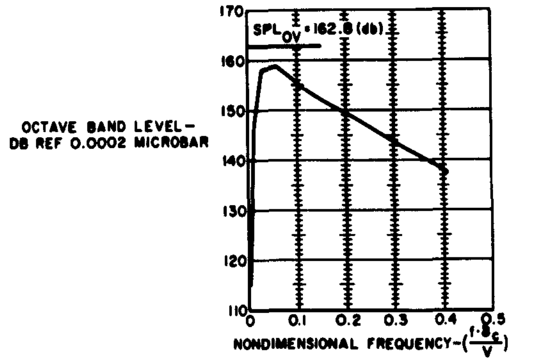


Fig. 6 - Nondimensionalized Little Joe 2 spectrum

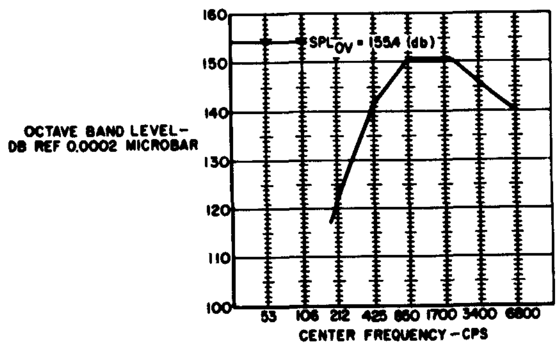


Fig. 7 - Boundary layer noise spectrum

SELECTION OF SIMULATION TECHNIQUE AND DESCRIPTION OF TEST FACILITY

The assembly to be tested is located inside a compartment where a reverberant condition prevails. Consequently, simulation of a reverberant noise field was the most suitable test method, and one of the reverberant test facilities designed by this company was selected for the test phase.

An external view of the test chamber is shown in Fig. 8. Air is supplied to four

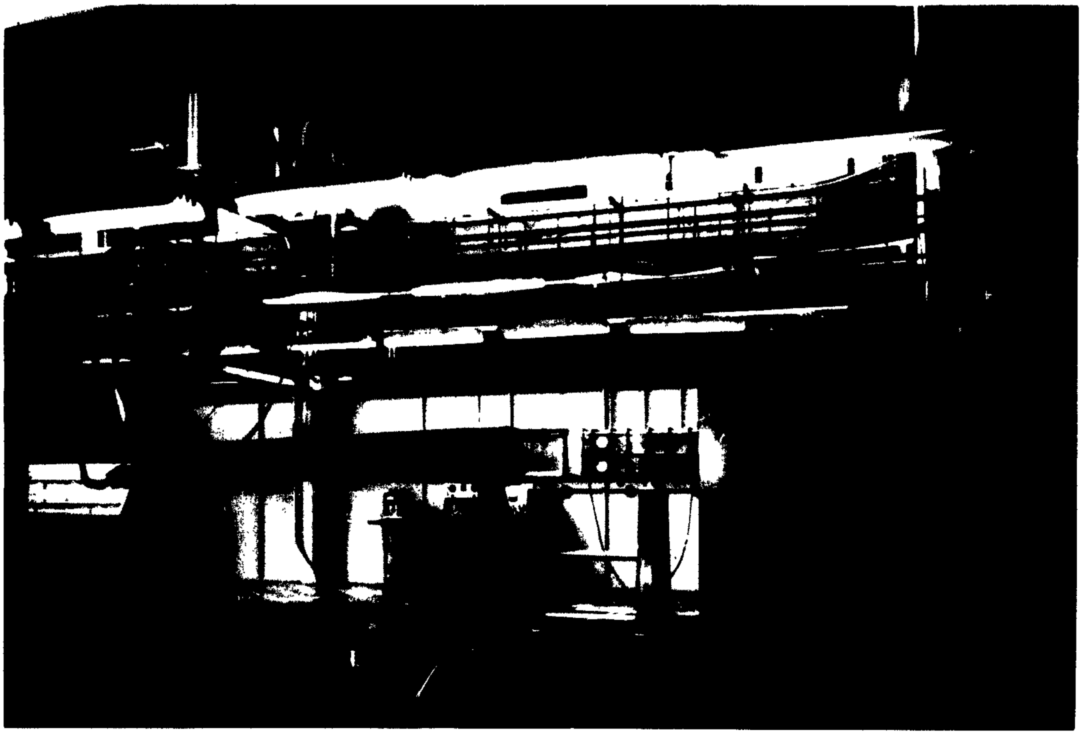


Fig. 8 - Bell reverberation test chamber, model 60D8B

electropneumatic transducers from a manifold of a regulated air supply. The transducers are coupled to the reverberation chamber (located within the sound proof housing) by an exponential horn, 23 feet long. The 13-foot section external to the housing, is covered with damping compound, fiberglass, and a plywood casing. This section is connected to the mouth of the second exponential horn by two rectangular chambers. The longer is a progressive wave test chamber. The other section is an acoustic mirror and mixer for adding high-frequency acoustic energy to the progressive chamber.

The reverberation chamber is irregularly shaped; it has 10 sides and a volume of 90 cubic feet (Fig. 9). There are 24 mid-frequency drivers, with exponential connectors, clustered about the low-frequency horn on 4 adjacent sides of the chamber. The illustration also shows an assembled section of the missile consisting of the radome, equipment, and payload compartments, and a portion of the engine casing, suspended in the chamber. The assembly tested is an arm safe device, consisting of many components, including electronic packages, electromechanical devices, transducers, and a barometric switch. These units were mounted on a one-piece magnesium casting. The

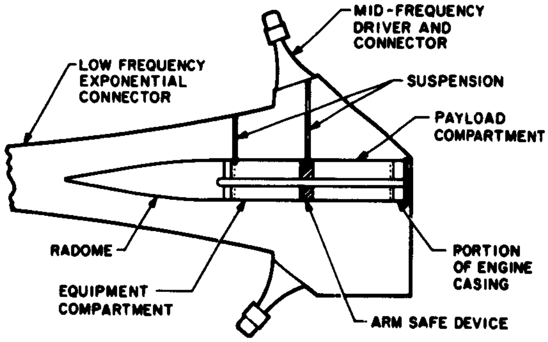


Fig. 9 - Suspension of assembled missile in the reverberation chamber

electronic subassemblies employed solid state circuitry and were well designed to withstand mechanical and acoustic vibration. To simulate the calculated noise fields, the spectral synthesizer shown in Fig. 10 was used. This system was designed to synthesize various types of missile spectra. Acoustic power for the low-frequency range is supplied by four electropneumatic transducers. Each transducer is driven by two preamplifiers and two 150-watt power amplifiers in parallel. The output from

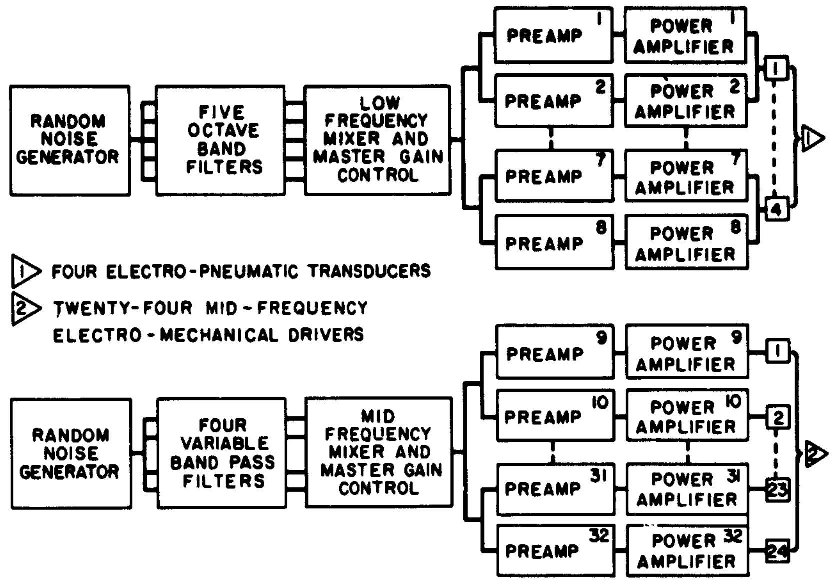


Fig. 10 - Block diagram, spectrum synthesizer, model 60D8B

a random noise generator is divided into five bands by octave bandpass filters, mixed in variable ratios, and fed to the eight preamplifiers through a master low-frequency gain control.

The acoustic power for the frequency range from 1200-10,000 cps is delivered by 24 high-intensity, mid-frequency, electromechanical drivers. Each driver is powered by one 150-watt amplifier and preamplifier. Similar electronics are used to drive the preamplifiers. Figure 11 shows a photograph of the synthesizer console. The random noise generators, filters, and power amplifiers are located in the vertical racks on both sides of the desk. The preamplifiers and overload warning lights are placed in the sloped section, and the mixer and master gain controls are located on the top of the desk. A sound level meter and octave band analyzer, located in the center above the desk, monitor the synthesized spectrum. At the right is an audio oscillator and frequency indicator for discrete frequency excitation.

The analysis indicated that the arm safe device, during ascent, is subjected to a superimposed noise field, originating from both the booster engine and boundary layer. This spectrum is shown by the solid line in Fig. 12. The internal spectrum, represented by the broken line in Fig. 12, was obtained from noise reduction measurements conducted with the assembled missile. During all tests, the internal spectra, originating either from both noise sources or from the boundary layer, were synthesized.

Two synthesized internal spectra, applied during a functional and a fatigue test, are compared (in Fig. 13) with the analytical spectrum. The differences for the upper four octave bands were within 1 db; while the four synthesized lower octave bands were 3 to 4 db high.

The operational tests of the arm safe device required simulation of the sequence of functions performed by the other missile subsystems initiated by certain flight parameters such as ambient static pressure and acceleration. The pressure altitude-time function was simulated by bleeding air from the baroswitch to an evacuated tank through a preset orifice. The acceleration-dependant actuators were replaced by solenoids, which were switched on at the same time intervals as would occur during actual flight.

TESTS RESULTS AND EVALUATION

The operational testing was performed in two tests. In the first, a so-called functional test, the arm safe device was exposed for 6 minutes to the superimposed simulated noise field, and was operated during the last minute. In a second so-called acoustical fatigue test, the exposure time was increased to 120 minutes. Again the device was operated only during the last minute. In both tests, the noise environment produced an output signal which, under actual flight conditions, would have caused destruction of the missile before reaching its intended target.

Fig. 11 - Spectrum synthesizer, model 60D8B

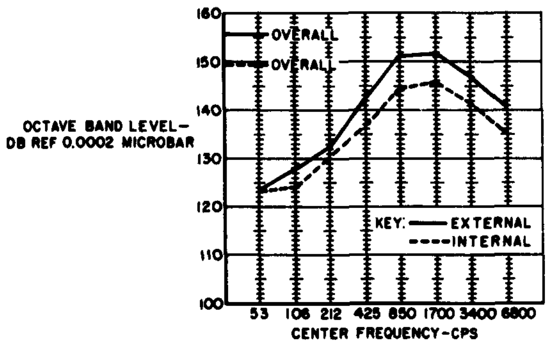


Fig. 12 - Spectrum from booster engine and boundary layer noise, superimposed

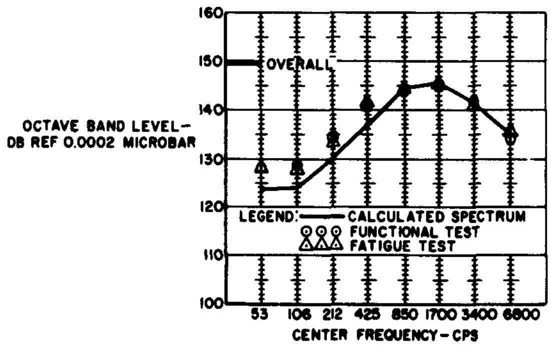


Fig. 13 - Synthesized noise spectra during functional and fatigue tests

Immediately after the tests, an operational bench test of the device inside the chamber without acoustical excitation did not show any deviation from normal operation.

An analysis of the recorded input and output functions revealed that baroswitch chatter caused the malfunction. Consequently, this component was among those selected for individual acoustic tests. The baroswitch alone proved to be sensitive to sound. This switch, shown schematically in Fig. 14, consists of four pressurized aneroid elements with internal contacts which open on ascent. The individual elements are connected in a series-parallel network. The baroswitch is set to activate at an altitude at which the missile operates at supersonic speed. Consequently, the baroswitch tests were

performed in a simulated boundary layer noise field, using either random octave band or discrete 1-cycle bandwidth spectra.

Baroswitch chatter was recorded on magnetic tape and played back for oscillograph recording. Excerpts from the oscillograms are shown in Fig. 15 for the network mostly closed (beginning of chatter), half open, and mostly open (end of chatter). A fundamental frequency of approximately 1600 cps is evident with a high percentage of second harmonic.

The chatter effect was also recorded on a magnetic tape loop machine and played back through a wave analyzer. The result, Fig. 16, shows the relative amplitude of the wave analyzer output as a function of frequency, for the network mostly open. The fundamental frequency again appears at 1600 cps and a second harmonic at 3200 cps.

Chatter duration dependence on pressure-altitude ascent rate is demonstrated in Fig. 17. The altitude error due to the chatter effect is the product of ascent rate and chatter duration. The error would be 1600 feet above the 7500-foot altitude setting. This graph also shows that introducing a time delay, to prevent chatter at minimum ascent rate, would result in an excessive altitude error at high ascent rates.

In subsequent tests, the four single switch elements and also the series-parallel network were investigated by the use of discrete frequency excitation. The results are demonstrated

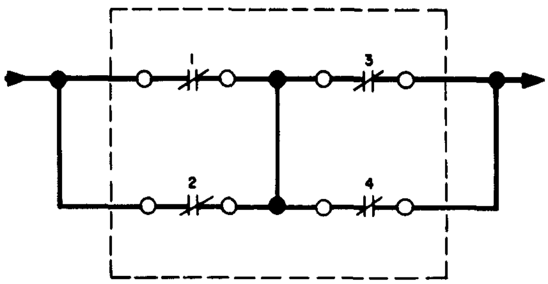


Fig. 14 - Schematic of baroswitch chatter

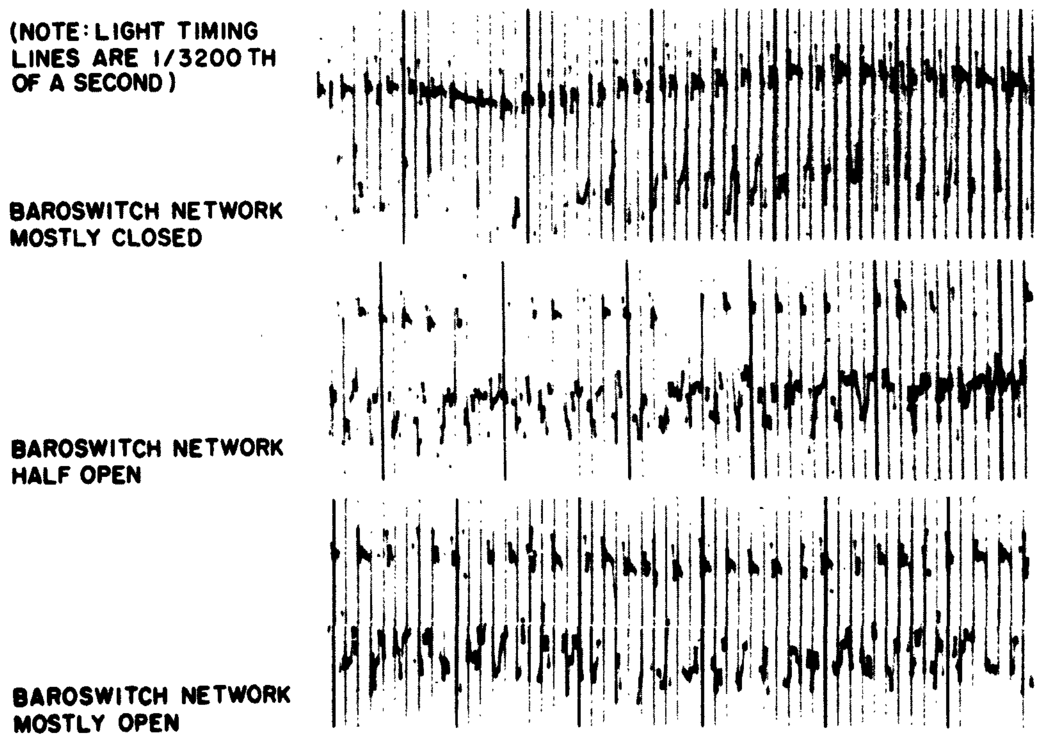


Fig. 15 - Oscillograms of baroswitch chatter

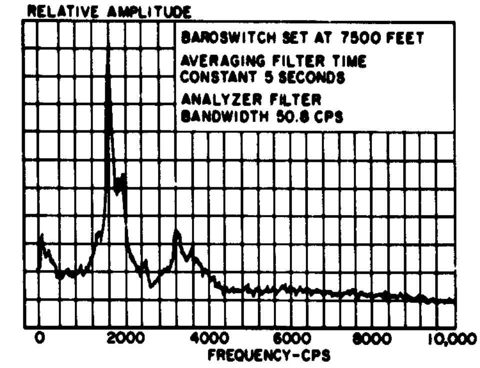


Fig. 16 - Wave analysis of baroswitch chatter with network mostly open

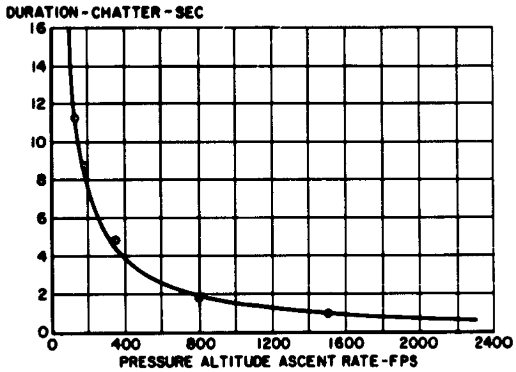


Fig. 17 - Baroswitch as a function of altitude ascent rate

in Fig. 18. The frequency bands and also the altitude ranges where chatter occurs, are narrow. Altitude chatter ranges vary from 35 to 370 feet even though the spectrum level peaked at only 117 db.

A chatter effect of the baroswitch network under discrete frequency excitation can be observed only if the chatter bands of two of the single switch elements, 1 or 2 and 3 or 4, coincide. In this baroswitch elements 1 and 4 have almost identical chatter bands, which produce network chatter above 1600 cps. The network also chattered at 1550 cps, but a chatter band in this frequency range was not found for either element 1 or 2. Very often a slight pulse is required to initiate a resonance which would then be sustained by a lower energy input.

The baroswitch setting was varied from 4500 to 65,000 feet of altitude and the fundamental resonant frequency was determined. Results are shown in Fig. 19. An increase of the bellows spring tension resulted in a decrease of the resonant frequency. A second mode of vibration was found between 4500 and 9000 feet.

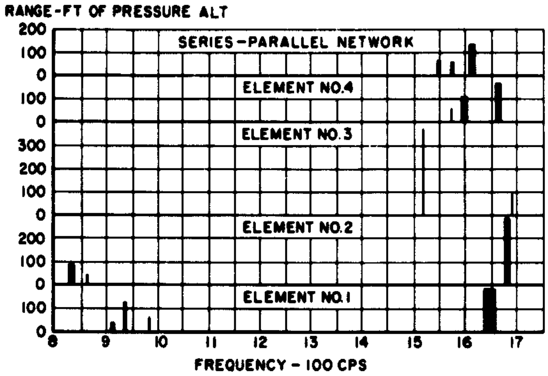


Fig. 18 - Chatter frequencies and ranges of baroswitch boundary layer spectrum levels (baroswitch setting at 7414 feet)

The range of altitude, during which chatter occurs, depends on the amount of excitation energy and is demonstrated in Fig. 20. Increase of the sound pressure level in increments of 5 db showed proportionality between pressure and chatter range.

The effect of the baroswitch chatter on the other components of the arm safe device was investigated in another test. This subsystem, subjected to random boundary layer noise at various ascent rates, showed the malfunction during each test. For slow ascent rates, the

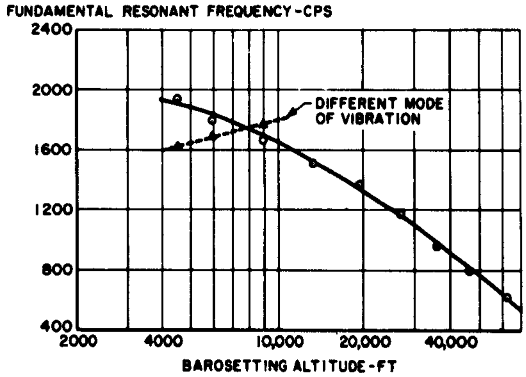
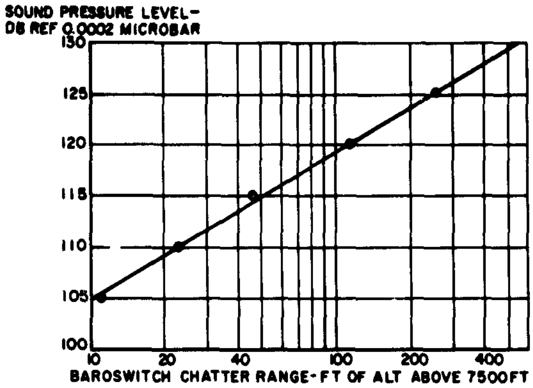


Fig. 19 - Fundamental resonant frequency of baroswitch as a function of barosetting



Fog. 20 - Altitude range of baroswitch chatter as a function of sound pressure level

malfunction occurred in one circuit path and for high rates in a second path. Intermediate rates resulted in a malfunction in both circuit paths. Based on the results of these tests, the circuit paths were modified and again subjected to operational tests. One modification resulted in improved performance, the other completely eliminated the effect of baroswitch chatter on the arm safe device.

CONCLUSIONS

The following conclusions may be made from the results of this test program. By the application of a simulated noise environment, the behavior of a sound sensitive component and its effects on a missile system can be analyzed. The results would enable a designer to perform an appropriate modification or a complete redesign. However, consideration of the acoustic environment in design and test specifications on a component level during the early design phase, would usually be more economical.

Appendix A

PROCEDURE FOR ESTIMATING BOOSTER ENGINE NOISE

Rocket engine experiments have shown that the acoustic power radiated from a rocket engine follows a law similar to that theoretically derived by Lighthill [2] for subsonic jets. This relationship may be expressed:

$$W = K\rho_\infty A V^\beta a_\infty^{-5}, \qquad \text{(A1)}$$

where,

A = nozzle exit area,

K = acoustic power coefficient,

W = acoustic power,

ρ_∞ = ambient density,

V = exhaust velocity, and

a_∞ = speed of sound.

The exponent β is found experimentally.

Acoustic power and sound pressure p are correlated by the equation of a spherical sound wave of radius r:

$$W = \frac{4\pi r^2 p^2}{\rho_\infty a_\infty}. \qquad \text{(A2)}$$

From Eqs. (A1) and (A2),

$$p^2 = \frac{M D^2 V^\beta}{r^2}, \qquad \text{(A3)}$$

where,

$$M = \frac{K\rho_\infty^2}{16 a_\infty^4}$$

and D = diameter of the nozzle exit.

For two noise sources that have different exit diameters D, the sound pressures at the distances r_1 and r_2 within the near noise field, respectively are:

$$p_1^2 = \frac{M D_1^2 V_1^\beta}{r_1^2} \quad \text{and} \quad p_2^2 = \frac{M D_2^2 V_2^\beta}{r_2^2}$$

for $V_1 = V_2$,

$$\frac{p_1}{p_2} = \frac{D_1 r_2}{D_2 r_1}. \qquad \text{(A4)}$$

If r is expressed in multiples of D_1, then:

$$r_1 = n_1 D_1, \qquad r_2 = n_2 D_2,$$

Or Eq. (A4) can be written:

$$\frac{p_1}{p_2} = \frac{n_2}{n_1}, \qquad \text{(A5)}$$

or

$$p_1 = p_2 \quad \text{for} \quad n_1 = n_2 . \qquad \text{(A6)}$$

Equation (A6) demonstrates that, for the same value of n and V, the same sound pressure can be expected.

Appendix B

PROCEDURE FOR CALCULATING THE LOCAL DYNAMIC PRESSURE

The calculation of the local dynamic pressure, at some point on the surface of a supersonic missile, requires that the total energy, entropy, and static pressure of the flow be known at that point. From the conservation of energy, the total energy (E) of a fluid particle at the boundary layer edge, must be equal to the sum of its thermal energy, under ambient conditions, and the kinetic energy imparted to it by the missile.

The entropy (S) of a fluid, as defined in thermodynamics, is a measure of the availability of the fluids energy in the performance of useful work. In an inviscid irrotational flow, the entropy will be constant along a streamline. For the case of a slender missile, such as the one under discussion, the ogive nose may be approximated by a conical surface, and the flow between the shock and the boundary layer will be irrotational as well as inviscid. Therefore, the entropy of the streamline at the boundary layer edge will be constant and equal to the entropy of the flow immediately aft of the shock.

The static pressure distribution along the body surface may be calculated from theoretical or approximate methods or obtained from experimental data. The theoretical values of the pressure coefficient (C_p) from Figs. 4 and 5 of Ref. 12 and Fig. 15 of Ref. 13 were used to compute the pressure distribution. These values of the pressure coefficient were obtained by using the method of characteristics described in Ref. 12 for a missile geometrically similar to the missile under discussion.

The detailed procedures used in determining the local dynamic pressure at the forward end of the missile compartment, are described in the following steps:

(1) At zero angle of attack and for a selected velocity (V_∞) and altitude from the trajectory information of Fig. 1, determine the ambient temperature (T_∞) and pressure (p_∞) from the atmospheric table of Ref. 13. The total energy (per unit mass) may be calculated from

$$E = c_p T_\infty + V_\infty^2/2 ,$$

where c_p is the specific heat of the ambient air at constant pressure. If T is the temperature at some point on the boundary layer edge, then the corresponding velocity (V) must be

$$V = [2(E - c_p T)]^{1/2}$$

(2) The entropy (per unit mass) of the flow behind the shock may be determined from the expression

$$S = c_p \log_e T_s - R \log_e p_s + S_o ,$$

where R is the gas constant for air, S_o is a reference entropy, and T_s and p_s are the static values of temperature and pressure behind the shock wave. The quantities T_s and p_s may be evaluated from oblique shock relations [14] in terms of the missile Mach number (M_∞) and the semi-nose angle (θ).

$$T_s/T_\infty = f_1(M_\infty, \theta) \qquad p_s/p = f_2(M_\infty, \theta).$$

With the entropy known, the temperature (T) at a particular point on the boundary layer can be calculated if the corresponding pressure (p) is known.

(3) The definition of pressure coefficient is

$$C_p = \frac{p - p_\infty}{q_\infty} ,$$

where p_∞ and q_∞ are the free-stream values of the static pressure and the dynamic pressure for the selected values of altitude and velocity of Step 1. With the pressure coefficient known, the pressure may be determined from

$$p = (C_p)(q_\infty) + p_\infty .$$

(4) With the pressure (p) known, the temperature (T) may be obtained from the entropy equation where S is the entropy from Step 2.

$$S = c_p \log_e T - R \log_e p + S_o .$$

(5) The local Mach number (M) may be calculated from

$$M = V/\sqrt{\gamma RT} \,,$$

where γ is the ratio of specific heats for air.

(6) Finally, the local dynamic pressure may be calculated from

$$q = \left(\frac{\gamma}{2}\right)(p)\,(M)^2 \,.$$

Appendix C

PROCEDURE FOR CALCULATING THE BOUNDARY LAYER THICKNESS

The incompressible turbulent boundary layer thickness (∂_i) at the compartment location, was calculated from the expression [15]

$$\partial_i = \frac{0.381\ L}{RN^{1/5}} \,, \tag{C1}$$

where L denotes a characteristic length, and RN the local Reynolds Number, which can be determined by the equation

$$RN = \frac{\rho VL}{\mu} \,, \tag{C2}$$

where ρ, V, and μ are the fluids density, velocity, and viscosity, respectively. This equation may be expressed in terms of the local dynamic pressure

$$RN = \frac{2qL}{V\mu} \,. \tag{C3}$$

It has been mentioned before that at low altitude and supersonic speed, the transition point is located only a short distance downstream of the vertex. We can, therefore, assume that the characteristic length L is equal to the distance between the vertex and the compartment.

The quantity V should be the average flow velocity along the distance L. We used the higher value calculated at the compartment. The error introduced by the larger value of L in Eq. (C1) is reduced to a negligible amount by the fifth root of the larger product VL in Eq. (C2). The coefficient of viscosity μ was calculated from Sutherland's equation [16]

$$\mu = 2.27 \times 10^{-8} \frac{T^{3/2}}{T + 198.6} \,. \tag{C4}$$

where T is the temperature at the edge of the boundary layer. Finally, the compressible boundary layer thickness ∂_c was obtained by multiplying the incompressible value of Eq. (C1) by the ratio ∂_c/∂_i. This ratio depends on the local Mach number and was taken from Ref. 15.

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Section 3

TEST TECHNIQUES

SYSTEMS EVALUATION AND THE TRADITIONAL ROLE OF THE TESTING LABORATORY

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General Electric Company

In this paper the system test will be identified and its position in the development picture discussed. Present needs of systems laboratories in terms of people, equipment, and support are also discussed and particular emphasis is placed on the greater freedom of test planning inherent at the systems level. Examples of inadequacies at the component level are given.

BACKGROUND AND DEFINITIONS

This paper must be read with the understanding that the opinions presented are those of the writer and they do not necessarily represent those of the General Electric Company or any of its employees. The ideas presented are the result of the writer's experiences in planning, conducting, and reporting on a variety of test programs for a number of government agencies and their contractors. Nothing of a classified or proprietary nature is discussed. It is hoped that the ideas presented may find reaction in similar thinking among engineers and managers working in this field and in more widespread use of similar ideas. Attempts will be made to keep the discussions general, but the conclusions drawn are necessarily affected by the writer's background.

It is important that certain terms be defined in the sense that they are used in this paper. These terms are:

- Component: This is the familiar black box, which is an assembly of piece parts (resistors, valves, transistors, regulators, lenses, etc.) designed to perform part of a subsystem function. Example: The guidance computer in an inertial guidance subsystem.

- Subsystem: An assembly of components which fulfills a discrete function in the successful fulfillment of the purpose of a system. Example: The inertial guidance subsystem of a ballistic missile.

- System: A major integral portion of a "Mission System" designed to enable that larger entity to fulfill a mission requirement. Examples: (1) The missile in a ballistic weapon system; (2) the spacecraft in an orbiting satellite system; (3) the AGE for the spacecraft; (4) the tracking system for an orbiting satellite system.

- Qualification: That portion of a testing program concerned with the demonstration of design compliance to procurement specifications (sometimes called Flight Proof, or Certification, or both).

INTRODUCTION

In the various institutions of learning which teach engineering, the student is exposed to

many ideals of his future profession. One of these is that concerning the partnership between design and experimental effort in the development of a product. In some industries, this partnership may still exist, but, in the aerospace and defense field, the contributions of the experimentalist to hardware development are too often ignored or wasted, or both. The traditional position of the experimentalist in the development cycle has been allowed to decay for the following reasons:

1. Up to recent times, the rapid advances in the state of the art in fields like vibration, acoustics, shock, space simulation, and thermal balance have consumed too much of the time of available experimental personnel. This has inhibited their ability to keep pace with test systems and hardware advances.

2. Principal development activity becomes centered on components at an early stage, thereby forcing a last-minute-aspect upon system level problem solution.

3. The hardware being tested has increased in complexity to the point where functional test equipment has become a separate problem; and the specialists work in this area alone. These special equipment design groups are usually divorced from the laboratories.

4. In many cases, the designer has shown a tendency to assume the test function, in spite of obvious objections to this practice.

Items 1, 3, and 4 in this list either are being corrected or have been corrected as time passes, but a change in approach is required for item 2. The national developmental effort continues to suffer the loss of the contributions of valuable technical personnel as long as this situation exists.

As long as the basic testing approach in product development remains component centered, with system problems considered as an afterthought, real contributions from the laboratory will remain, "Too little and too late."

It is the purpose of this paper to explore a simple logical expansion of the ordinary systems testing effort in a manner devised to capitalize upon the unique capabilities of experimental personnel. The paper begins by briefly describing a typical development cycle, exploring its limitations, and indicating the prevalent waste of valuable talent. The increased potential in a system centered effort is shown to be appreciable. For the purposes of this paper, the vital Quality Control testing activities will be omitted from discussion.

PRESENT APPROACH

A key factor in most space and defense contracts is the lack of time. Money is usually available, but development schedules must be compressed continually in the interests of meeting critical delivery dates. The concept of a design-experimental team activity in such circumstances is lost in the race to deliver. The familiar design-test-redesign cycle of hardware maturation is destroyed and great emphasis must be placed on the design function because the test contribution becomes one of demonstration rather than development. This time deficiency is difficult to overcome at the component level unless a determination can be made as to where component test emphasis should be placed. In the vital systems area, some relief is available, if experimental capabilities can be utilized earlier in the effort. In the present development cycle, this is not being done. The following steps usually constitute the present development cycle:

Conceptual Design—This is a pure paper-and-pencil effort where the experimental personnel are usually omitted unless the customer demands a comprehensive overall test plan. In most cases, the valuable first-hand knowledge of recently completed tests, on the previous generation of hardware, is either lost or not applied until later. The design effort starts at a system level, but it is rapidly broken down to the component level to permit large numbers of designers to work on the problem. This somewhat disjointed design period is where system-oriented experimental personnel, if given the opportunity, could be most profitably employed.

Design—The concepts are reduced in this stage to drawings and the experimental people begin to contribute test plans, buy facilities, and design special test equipment. This latter equipment is inevitably behind schedule and a panic party develops because it was not started earlier. This unnecessary situation could be avoided if the experimental personnel could be integrated into the program earlier. The major testing effort is centered at the component level and it is usually limited to functional considerations, with the services of the testing laboratories available to the component designer. The key testing personnel are still marking time, waiting for the day when their turn will come. At this time, systems evalatuon is usually deferred until component problems have

been resolved. This can be a fatal error for system bugs often more difficult to overcome.

Component Evaluation—At this stage, environmental tests are injected into the cycle and, as a more formal evaluation effort gets underway, serious attempts are made in the laboratory to adhere to a specification. Because of the component-centered approach and the strong emphasis on structural design, this latter design area is often emphasized at the expense of other areas where the predicted failure rate is higher. Modern practice has demonstrated, on many types of contract, that the black box subsystems cause more failures in service than the basic structure. If systems evaluations are planned, the senior experimental planning personnel begin to enter the program. These people are not usually involved in the component efforts, because of the prevalent practice of using the laboratory as a service shop. If component-level qualification tests are required in this program, they can be started at this point for those components which do not have serious development problems. Meanwhile, much time has been wasted on functional overdevelopment at the component level. Functional specification requirements are often too stringent and "Cadillac" components are developed when "Chevrolet" hardware would do the job.

Evaluation—It is during this phase that long duration programs benefit from the senior laboratory personnel because they begin to contribute by conducting an independent evaluation of the equipment. There is no such luxury on rush programs where this effort may be omitted entirely in progressing to the demonstration tests described next. One complication in this evaluation effort is the probable existence of a design freeze which inhibits most attempts at design improvement. The best efforts of the experimental personnel are either lost or, at least, sharply reduced in effectiveness by such a condition. The other bad feature about this phase of the cycle is the questionable condition of the hardware available for test. A test system often resembles a patchwork quilt of hardware vintages and conditions. The specimen integrity is such that test results are often questioned for this factor alone.

Demonstration—This is often the end of the line for the engineering function with Quality Control assuming responsibility for tests beyond this point in the program. The laboratory is expected to produce the failure-free completion of a systems level test program devised to demonstrate design compliance. The state of the art, in this area, is such that assurance of design compliance often cannot be provided with tests of a single system. The complexity of the environment has created such problems as equipment wearout on long tests, the need for marginal explorations (step-stress testing), the need to compress time schedules, and the need for specific tests where relatively crude models must be used (antenna tests). More crucial time is consumed at this stage in correcting mutual compatibility problems, which should have been cleared up earlier in the development effort. In some programs, work with succeeding modifications must be done with this same overtested system. Such test programs often must be devised on the basis of inadequate acquaintance with the hardware. The risk of failure of the test equipment or test method, or both, is often higher than the risk of test hardware failure. Without benefit of early first-hand acquaintance with the hardware, and with inadequate preparation for inter-component problems, the demonstration is often disappointing.

All of this presents a picture of well-staffed experimental laboratories becoming involved in testing programs, only after it is too late to best use their recommendations in the improvement of the final product. This gross waste of a segment of the available skilled technical manpower should be stopped. There is a method of operation which will permit this manpower to function in the development cycle in a timely manner.

PROPOSAL

The basic idea in what is called the Experimental System Analog Approach, is to place what is sometimes used as a design tool in the hands of experimental personnel where it can be of greater use. Most of the actual work on such an analog is experimental rather than deliberative and, as such, is entirely different from that usually required from a designer. When designers use such a tool exclusively, much of its value is lost in the limited scope of its application.

It is proposed that a complete system analog be established, as early as possible in the development cycle, in the laboratory areas. Features of this rather special analog must be:

- Ease of substitution, component for component, as real hardware becomes available in the program.

- The ability to segregate a component, in operative condition, to subject it to a variety of environmental stresses.

- The ability to install the entire analog in the larger test facilities when preliminary system evaluations are in process.

- Extensive instrumentation for readout.

- Capability for parametric investigation on component performance as well as simulated responses.

- Adaptability to program changes.

Such an approach should be confined, initially, to programs where the scope can justify this additional sophistication, but future utility can be broadened. It is about time that the laboratory areas capitalized on the versatility of computer approaches in areas other than data processing and automated Go - No Go test control. The extreme flexibility of the analog can broaden the experimental capability without unusual magnification of experimental costs. In the long run, savings may be realized by the use of these methods.

A gross advantage which cannot be overemphasized is the restoration of the independent system level evaluation, in a timely manner, on programs where it is really needed. The experimental personnel will be able to explore the systems concepts when something has been done about the problem areas in mutual compatability, and so on.

NEW OPERATION

This technical specialty, which is sometimes called the environmental field, has been beset with a number of magic words in recent years, the ultimate reaction to which has been detrimental. Words like random vibration, reliability, human factors, and maintainability have served some initial purpose, but severe abuse of these terms has detracted from their significance. It seems that the latest of these is "systems test." It is indeed regretable that such a powerful tool of evaluation should have to develop in such an atmosphere. Most engineers would like to see some genuine technical benefit occur from the use of systems level techniques without the accompanying fund raising ballyhoo. In view of the session devoted to this subject at this 31st Symposium and articles and editorials in recent publications, one might ask "What is all of the fuss about systems testing?" Many laboratories have been running tests on a systems level for years without any special notice of it. There must be some special additional gain to be realized in this area. The real gain is linked to the improved performance which is available with full exploration of a systems altitude, and attitude is essential to success in this effort. In the specifications area alone, it must be realized that systems-level shock, vibration, acoustic, temperature, and vacuum test specifications can be much more realistic than comparable component-level specifications. This is particularly true at an early development stage. It is not possible to investigate compatability problems on a better basis and electromagnetic interference testing (EMI) is virtually meaningless on the component-level. In many ways, a better job of evaluation can be done at the systems level and the experimental analog enhances the ability to work at this level in a timely manner. The following paragraphs describe how the Experimental System Analog Approach can show real product improvement contributions on a development program in a manner now largely ignored:

Conceptual Design

The systems level foundation work is translated into a computer-based analog for the study of parametric responses as soon as component breakout occurs. If known sets of hardware from earlier programs are to be used, the actual hardware can be inserted in the analog. This analog should be used in the laboratory by experimental personnel in order to exploit fully their intuitive understanding of the experimental method. The use of a purely functional analog by design personnel has, in the writer's opinion, produced inadequate gains in the recent past. The use of dynamic analogs for vibration problem areas has done little to enable the designer to overcome his basic limitations when making assumptions in this field. The original assumptions, in most cases, cannot be modified easily as the program matures. As a design tool, this invaluable analog equipment exhausts its usefulness early, and much of its ultimate value can be lost because of this. It is the writer's conviction that this demonstration has been the result of a need for investigative rather than deliberative experience on such equipment. Feedback from this analog should enter the design area continually and in a timely manner. This analog also will serve as a means to train evaluation personnel to permit more independent functional evaluations where they are required. (In the normal program, the designer must accompany the equipment into the laboratory because no trained personnel pool exists.)

Design

As component definition improves, more and more hardware will replace computer

elements in the system analog. System parameters will become more meaningful as the assumptions become more realistic. More and more reactive feedback between the design and experimental organizations will provide mutual benefit in the form of trained personnel, better hardware and better preparation to test the hardware when it is required to do so.

Component Evaluation

This entire effort will become more meaningful because efforts can be concentrated where they are needed. Problems of interaction, and so on, will have been resolved on the, now, semi-analog, and the analog can be used as a tool for greater realism when the components are tested. Thermal test results, which are questionable under some present programs, can be made more realistic and the results can be much more useful.

Evaluation

Here, concentrated early efforts can show some results, and the analog will come into its own as a testing tool. Experience in its use will have sharpened the ability to make the assumptions and the analog will make an excellent response stimulator for the system under test. Feedback from inertial and aerodynamic sources will be available, when it is required, for the exercise of the hardware under test environments. Instead of using vintage hardware, the analog components may be updated by parameter adjustment with improved tests of the remaining real hardware.

Demonstration

The improved ability to exercise a completed system will become an asset in the demonstrative test phases. A wider range of stimuli are to be expected and parametric extremes can be used in the tests. An additional advantage lies in the further development of the system by means of the analog. Component performance parameters can be changed conveniently as well as the various stimuli. With the actual test hardware, this is often difficult. It is even possible to accelerate the test effort by making it possible to finish a test run with the analog component in place of a failed one. This can enable the program to salvage manhours, materials, and time which would be lost in a major test shutdown. The greatly reduced failure risk will result from the improved knowledge status in the experimental areas.

Key Advantages

The brief discussion presented herein should provide an insight into the potential of a correctly applied Experimental Systems Analog approach. Much more can be said when a specific problem is discussed. The generalized problem should point out the key advantages, which are:

- Better use of skilled evaluation personnel in the development of better hardware.

- Superior test methods because of more complete parametric investigations.

- Better environmental simulation in key areas like vibration and thermal testing. This is based upon the innate ability to write better systems specifications.

- The possibility of saving accumulated test hours when a single component failure occurs.

SUMMARY

A very brief discussion has been presented on the use of an Experimental System Analog in the early phases of a hardware development program. This is presented in terms of a positive suggestion and in no sense should the test effort be used to replace sound design. No amount of evaluation can, by itself, create better hardware. The testing does provide a vital form of feedback which has deteriorated in present day rush programs. It is the writers contention that the feedback loop should be reshaped and reclosed. The alternative is the present situation where test results on Program A can only effect the hardware of Program B. (In too many cases Program B is in the hands of another contractor and the experience gained from Program A is never fully utilized.)

In order to accomplish the objectives claimed for the Experimental Systems Analog approach, the laboratory of today will have to change somewhat. The computer must become a vital tool in such a laboratory and it will do much of the routine work. Properly used, it is possible, now, to have routine service tests run by simple computers having preset failure parameters and corrective programs. The manager of such a laboratory will have to broaden his hiring base to include personnel capable of designing, constructing and utilizing the systems analog. The average engineer in such a laboratory will become versed in the functional

aspects of a system under test as well as the environmental parameters to which the system will be exposed. This new breed of evaluator will restore the ability of the laboratory to contribute to hardware development by the use of the same tool which has freed the conceptual designers hands for more advanced effort.

The costs of experimental systems analog programs, like that described previously, will restrict their initial use to larger programs where the added cost can be justified. When the basic equipment becomes available in a given laboratory, it will be possible to adapt it to a wider range of test programs.

DISCUSSION

Mr. R. Jones (Admiral): You gave the impression in your talk, and I think I have a tendency to agree with you, that designers are very untrustworthy hobbyists.

Mr. Yaeger: No. I'm sorry if I did give that impression. The only thing I feel is that it is unwise to have any designer, including myself, evaluate his own hardware. I think this is a highly unsatisfactory condition. I also feel that we should attempt to turn experimental work, in the case of the use of an analog, over to people who by nature are experimental rather than deliberative in their basic work. The designer deliberates or decides between various objectives and various things that he intends to use, while the laboratory person must try things. I think that if perhaps we tried this experimental technique once with the use of a systems analog, that we might get greater benefits from the analog. I have a great deal of respect for designers. I know quite a few of them and I've tried to get along with them as well as I can.

Mr. Jones: Are you not, by following this approach, simply taking this hobbyist facility away from the designer and putting it somewhere else?

Mr. Yaeger: In a sense, yes, because I'm firmly convinced we can do more with it there. That doesn't necessarily mean that we should throw the designer's analog out, but I think his basic utility of it is exhausted at too early a phase in the program. This is the whole point. I think we should carry this into the testing laboratory because I think it can be of tremendous use to us there.

Mr. I. Sandler (Autonetics): I don't have a good feeling for the analog experimental or the designer analog as of right now. I wish you would clarify this.

Mr. Yaeger: Well, usually the designer's use of an analog is restricted to functional considerations. He uses it as a tool to compare performance parameters of individual components. He has something which acts like a guidance computer but which isn't actually a guidance computer. It takes the inputs, let's say it simulates the responses of a guidance computer to the inputs which a guidance computer gets.

Mr. Sandler: Well would this be similar to an analog computer which is taking the place of each component and you integrate the analog component?

Mr. Yaeger: In a sense it might, I say analog, actually I'm too restrictive in the term. It might well be that you would use digital components for certain items.

Dr. Irwin, Chairman (NRL): Could I ask a question? This may be naive, but is there some difference between what you mean by an analog and what we used to call a breadboard model?

Mr. Yaeger: There can be quite a bit. We would place provisions into the analog so that we could simulate things like aerodynamic and inertial responses of reactive components to the environment. When one is working on a system-level he must consider this. If it is a breadboard of a given component, it is usually electrical input to electrical output.

J. Bakalish (Martin Denver): You spoke of a deliberative effort. Have you considered putting this deliberative effort in someone elses rather than just the designers hands?

Mr. Yaeger: No, that's his job. His job is essentially making decisions between components and what he intends to put in the system when he designs it.

Mr. Bakalish: Well, I was thinking from the standpoint of deliberative efforts in analyzing his problem and coming up with possible data that would lead to a better conclusion.

Mr. Yaeger: If you are talking about analysis or design problems, I think these are pretty

much his. If you are talking about experimentally investigating the equipment he comes up with to determine if it does the job satisfactorily or not, that's basically the testing laboratory's problem. Perhaps I've misunderstood your question.

Mr. Bakalish: I was thinking of an instance where you have a problem within the overall component, in which the component itself may do the job, but you are having a problem with a subcomponent or part of the design.

Mr. Yaeger: The problem in this sense, we touched on it this morning, is that in developing systems, particularly at the rapid rate we have to do nowadays, we find out that some of the hardware is available the day the system is designed because we've used it on the last previous set of hardware of this type. A lot of it is not available sometimes until the day we deliver it, as far as testing it is concerned. Our whole testing effort sometimes is slowed down and geared down because we have no means of testing a lot of the equipment. It simply just is not there. Whereas, if we had something, in the laboratory, which resembled a system and would enable us to test the hardware that is available, then we could step the whole thing down stream somewhat, at least get the testing effort off the ground early enough in the program so that we could profit from it.

D. Sing (Bell Aerosystems): I would like to defend the designer. Unfortunately, I think it is the designer's responsibility whether or not the thing works, not the experimenter's. There is one other comment I have to make—there is such a thing as a systems engineer. Now is not this his responsibility rather than that of the test department?

Mr. Yaeger: This varies from organization to organization. When you use a semantic term like system engineer, I know what this means right now at GE, I work for one, but what this may mean at Bell Aircraft nowadays I don't know. The one I work for has chiefly a coordinating function. He does not actually do any direct work. He or his people run around and get answers from other people about what other people are deciding to do.

Mr. Newhouse (Marquardt): But isn't this really an organizational problem that you are talking about? We work as a development team which, I would gather, you classify as experimental with designer right on the board as a member of the team along with the analysis group. As we work together past experience is brought right in on the initial design.

Mr. Yaeger: This is splendid. I haven't seen too many organizations to do this. More power to Marquardt.

* * *

AN APPROACH TO POLARIS FLIGHT SHOCK SIMULATION BY ELECTRODYNAMIC SHAKER*

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This paper describes the initial use of electrodynamic shakers to simulate black box transients in the POLARIS flight environment. The results indicate the measure of success in generating arbitrary pulse shapes. The shaker responses obtained by photographic time-plots were free of distortion and were repeatable in all aspects.

INTRODUCTION

Several approaches are used for simulating the shock environment experienced by missile structure. To date, the state of the art has been limited primarily to test simulations by electrodynamic shakers or impact devices using singular positive pulse function inputs such as half-sine, triangle, sawtooth, and others.[1,2] The question may be asked: are these shock techniques satisfactory from the standpoint of realistic simulation of a missile's ground and flight transient environment? The answer to this question must be stated in terms of the degree of expected environmental fidelity.

The test approach to be discussed consists of the application of arbitrary shock waveforms as input to an electrodynamic shaker system. These shock waveforms realistically described the POLARIS missile ground or flight environment. This approach has several pronounced advantages over present test methods. These advantages are enumerated as follows:

1. Stresses are reproduced on a time-dependent basis similar in character to the stresses incurred during logistic, launch, and flight patterns. These stress cycles, considering functional modes of failure, establish a realistic test concept and the singular positive pulse shapes do not. The question of whether this failure would have occurred during actual flight environment, always arises in the event of a functional failure under singular-pulse test conditions. In other words, functional failures may occur under singular-pulse test conditions, but not during actual flight environment, and vice versa. To minimize any doubts of this nature from test to test, primary importance was given to the distortion characteristics between the shaker's input and response functions, stability of the function generator, and repeatability of the intended real-time acceleration.

2. The stored energies, inherent in the simulated arbitrary shock waveform and the singular positive shock pulse, are dissimilar in their dissipation.

3. Selection of the proper half-sine pulse or similar singular pulses as an envelope of a given shock spectrum generally compromises the system. Adjustments must be made in fitting these curves which inherently cause missile components to be either severely under- or over-tested in particular frequency bands. The optimization of the shock test requirements by this approach has an important impact on the development and reliability of missile equipment.

*Performed under Bureau of Naval Weapons Contract NOrd 17107.

[1] Lewis, H. O., "Shock Testing POLARIS Missile Re-entry Bodies with an Electrodynamic Shaker," Shock, Vibration and Associated Environments, Bulletin No. 28, Part IV (Aug. 1960).

[2] Wells, R. H. and Mauer, R. C., "Shock Testing with the Electrodynamic Shaker," Shock, Vibration and Associated Environments, Bulletin No. 29, Part IV (June 1961).

SHOCK SIMULATION SYSTEM

The shaker system used for the initial series of tests consisted of a Ling Model 177A, 5000-force-pound shaker and an appropriate electronic power amplifier. This system was designed to operate in a frequency range of from 20- to 2000-cps. The shaker armature weighs approximately 92 pounds.

The compensating network required for the correction of shaker resonance and phase angle was made up of capacitance and resistance decade boxes. More sophisticated compensating networks could have been selected for these tests. It was felt, however, that for proving the feasibility of shock simulation, simpler black boxes were adequate.

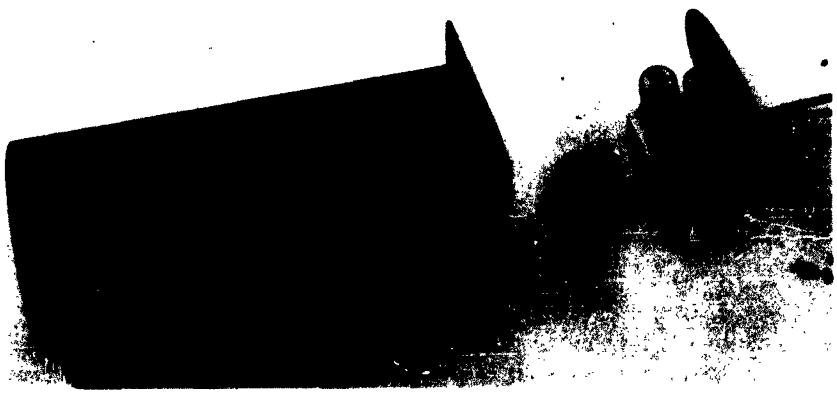
A photocell function reproduction technique was selected as the most proficient means for electronically reproducing the required complex wave pulse. This technique of reproduction is not unique. It was initially conceived as a function generator for analog computer work. Even its use in conjunction with shock simulation is not new.[2] However, in designing and checking out this new photocell function generator (Figs. 1 and 2), particular attention was given to overcoming the unfavorable aspect of controlling the electron beam. A detailed description of the photocell function generator may be found in the Appendix.

TEST PROGRAM

The test program designed to establish the feasibility of this method of shock simulation was simple, yet comprehensive in all aspects. A progressive step-by-step approach was adopted to ensure continued control over the experiment.

Three conditions of shaker loading were applied: bare shaker table, 20 pounds of rigid

(a) - Closeup of photocell chassis



(b) - Photocell with shroud

Fig. 1 - Photocell assembly

body mass, 41 pounds of fixture including elastic mass attachments, and 34 pounds supplementary rigid mass. Duration, level, and pulse forms were monitored by plotting the output of two accelerometers on an oscilloscope. One accelerometer was mounted on the shaker table and the other on the rigid mass or fixture. The resulting traces were recorded by a Polaroid Land Camera. This method of recording was felt to be sufficient for initial data analysis of input and response shock simulation traces.

Three typical POLARIS in-flight transient pulse shapes were selected for the shake tests. These pulse shapes were selected on the basis of their diverse characteristics in durations, frequency content, and differential acceleration amplitudes. These shock waveforms were intended to cover the broad spectrum of POLARIS missile shock history. Figure 3 shows the acceleration-time plots of these three model shocks. Figure 4 illustrates, in detail, the instrumentation and equipment used in this shock simulation study.

TEST PROCEDURE

To obtain repeatable as well as properly shaped shock data in the test laboratory, stringent control must be exerted on all key equipment involved in the test setup. A method for

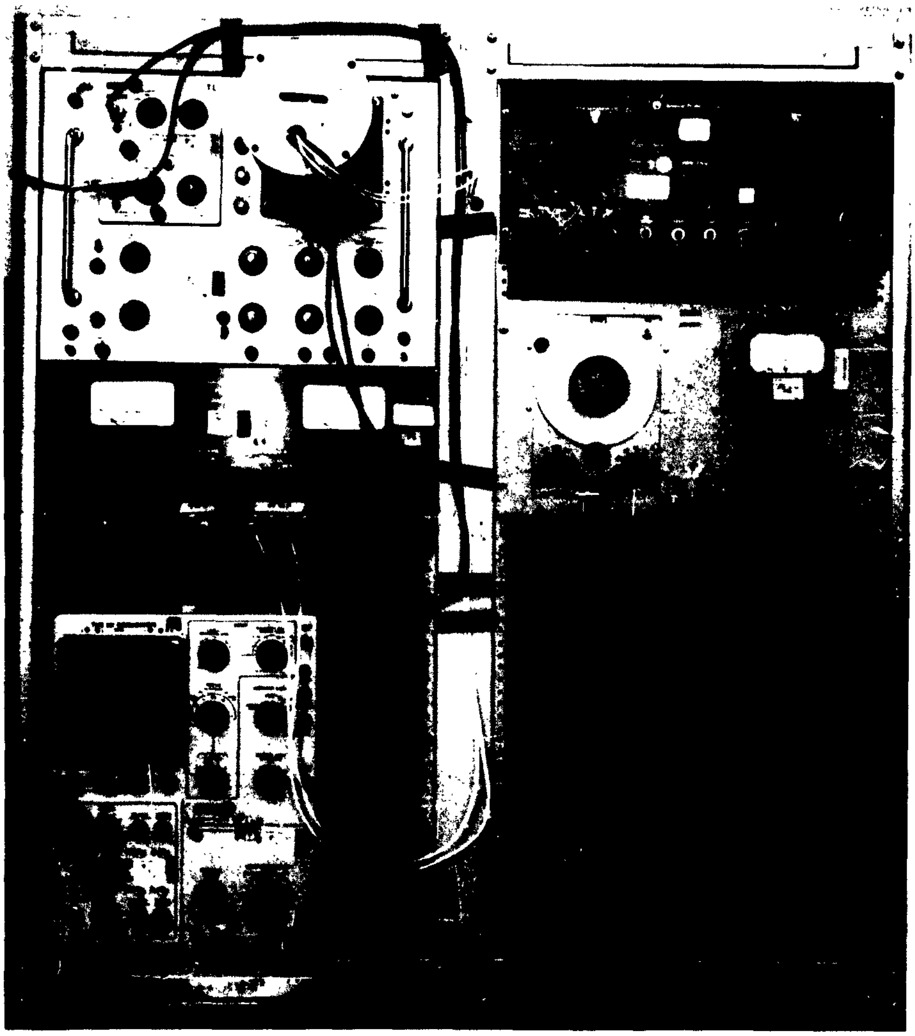


Fig. 2 - Photocell function generator instrumentation

controlling the characteristics of the electrical analog input function is essential. This is necessary so that exciter response, in the form of table acceleration, can be shaped to the desired waveform.

If the vibration exciter system is considered as an open-loop system, the acceleration output of the shaker is the controlled variable and the input electrical analog function the reference input. The transfer function characteristics inherent in the system must be evaluated and corrected. This must be performed before any definite regulation of the controlled variable or shaker table acceleration waveform can be accomplished. In the case of vibration exciters, transfer function correction requirements change with both shock pulse frequency and table loading. During this test program, the method employed for compensation or transfer function correction consisted of building-up an electrical analog compensating network. The network, as part of the loop, was installed between the electrical analog function generator and the master gain control at the input to the exciter power amplifier. The compensation

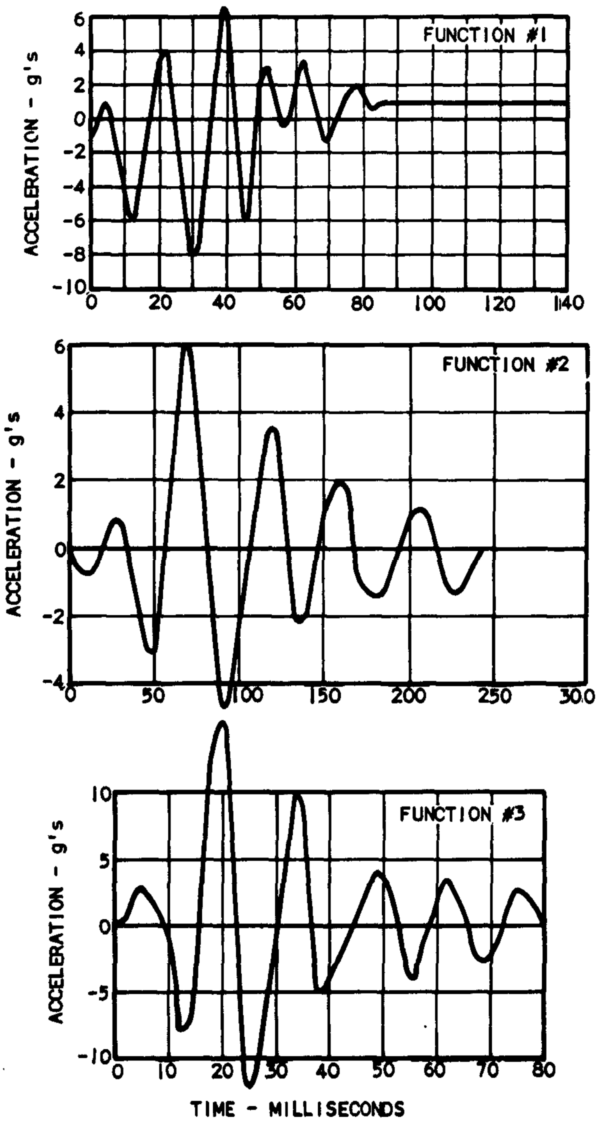


Fig. 3 - Model shocks

network was completely flexible and would compensate through a broad dynamic range.

Evaluation of this compensation network, with a simulated shaker in electrical analog form, showed that its compensation-range-coverage exceeded the most severe exciter load condition which could conceivably be experienced from the three shock models. The network had the capability of equalizing or compensating for poor exciter response and was capable of providing electrical damping and phase angle corrections.

For practical reasons, the transfer function necessary to compensate the power amplifier input signal was determined experimentally at a low acceleration level. This was accomplished by comparing the graphic presentation of the prescribed pulse shape and the resulting uncompensated shaker response transient.

After the first attempt at compensation had been made, a second shock pulse was generated and superimposed over the initial trial pulse. This process can be repeated as often as necessary to allow for the highest degree of satisfactory compensation. Once the waveform was cleared of distortion, the power amplifier gain was raised to the predetermined test level.

TEST RESULTS

The test records (Figs. 5-26) derived from the two or more accelerometers, are arranged in order of test sequence as the shaker-loads were increased in severity: bare table, 20-pound rigid mass, 41-pound test fixture (including some elastic masses), and 34-pound rigid mass. For the study of distortion characteristics and determination of compensation settings, uncompensated traces were obtained for each shaker load and prescribed transient function combination. Also in some instances, the sweep rate of the function generator was varied for some of the combinations to assure the best resolution of the point trace. To date, none of the response data has been recorded on tape. It was felt that the photo recording technique would be adequate to establish the feasibility of this shock simulation.

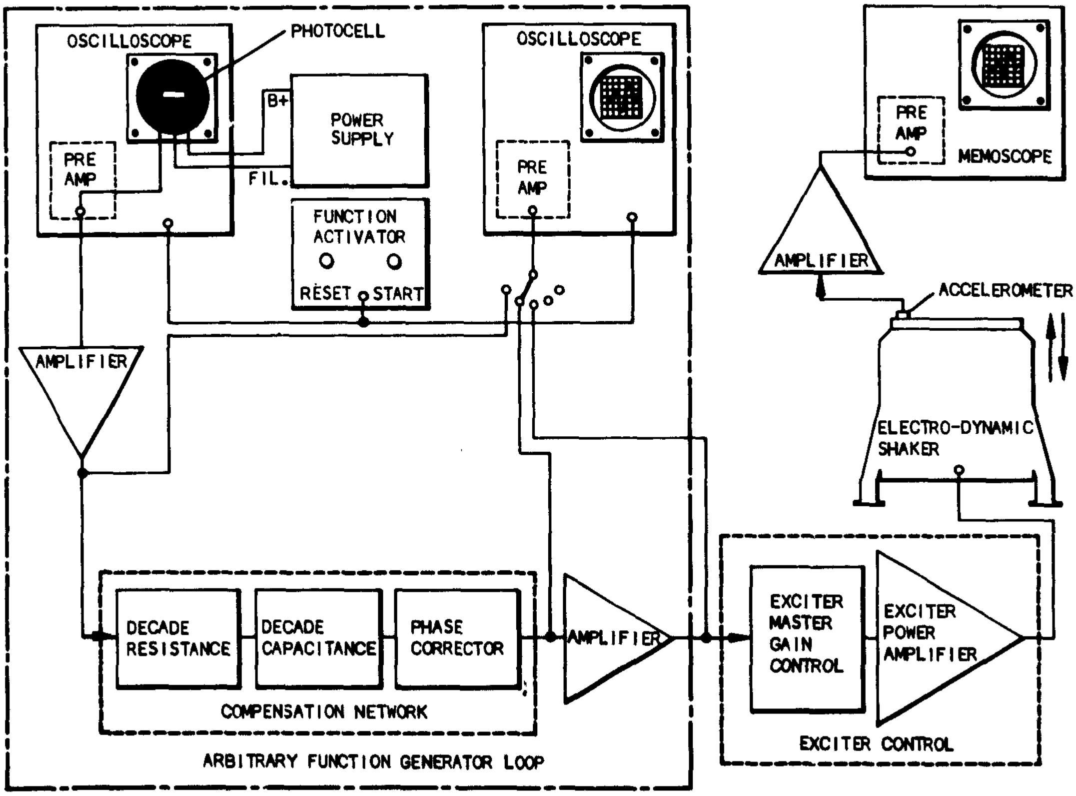
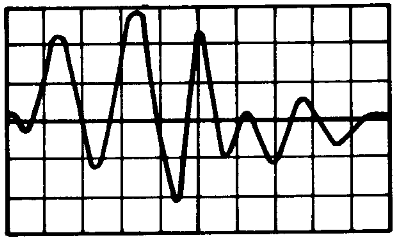
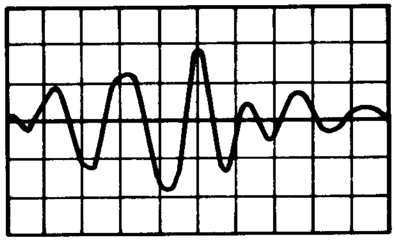


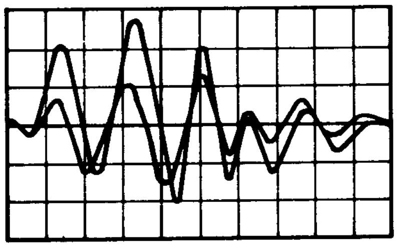
Fig. 4 - Block diagram of system



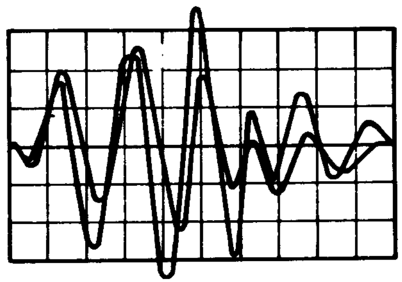
A. FUNCTION GENERATOR OUTPUT VOLTAGE
SWEEP: 85-MILLISEC
VERTICAL: 6-VOLTS FULL SCALE (1 VOLT/DIV)



B. SHAKER RESPONSE - ACCELERATION
SWEEP: 85-MILLISEC
VERTICAL: 3-VOLTS FULL SCALE (15 V/DIV)

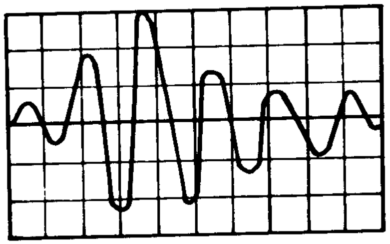


C. SIMULTANEOUS DISPLAY OF FUNCTION GENERATOR OUTPUT AND SHAKER RESPONSE
SWEEP: 85-MILLISEC
VERTICAL: FUNCTION GENERATOR OUTPUT - SAME AS A. SHAKER RESPONSE - SAME AS B.

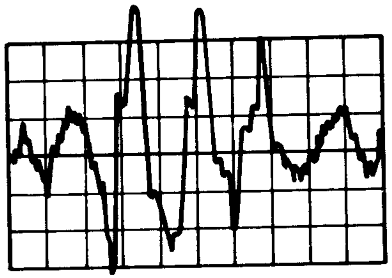


D. SIMULTANEOUS DISPLAY OF FUNCTION GENERATOR OUTPUT AND SHAKER RESPONSE
SWEEP: 85-MILLISEC
VERTICAL: FUNCTION GENERATOR OUTPUT - SAME AS A. SHAKER RESPONSE - 1.2 VOLTS FULL SCALE.

Fig. 5 - Bare table response characteristics of 5000-force-pound exciter to function No. 1 without input-signal compensation or shaker equalization



A. FUNCTION GENERATOR OUTPUT VOLTAGE
SWEEP: 240-MILLISEC
VERTICAL: 1.2-VOLTS FULL SCALE (.2 VOLTS/DIV)

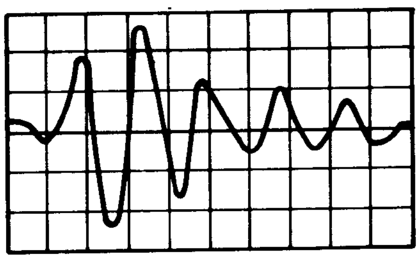


B. SHAKER RESPONSE - ACCELERATION
SWEEP: 240-MILLISEC
VERTICAL: 1.2-VOLTS FULL SCALE (.2-VOLTS/DIV)

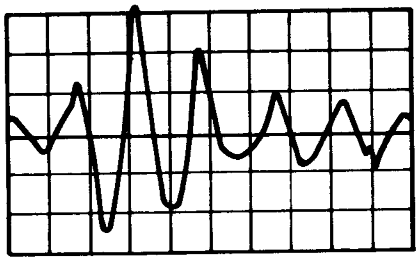
C. SIMULTANEOUS DISPLAY OF FUNCTION GENERATOR OUTPUT AND SHAKER RESPONSE
SWEEP: 240-MILLISEC
VERTICAL: FUNCTION GENERATOR OUTPUT - SAME AS A. SHAKER RESPONSE - SAME AS B

D. REPEAT SHOT WITH SAME CONDITIONS AS C TO ILLUSTRATE REPEATIBILITY OF DATA.

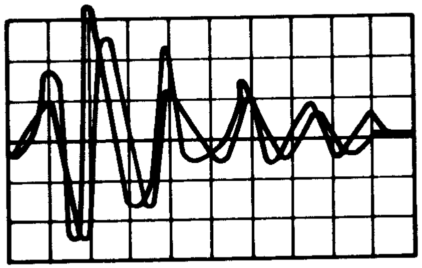
Fig. 6 - Bare table response characteristics of 5000-force-pound exciter to function No. 2 without input-signal compensation or shaker equalization



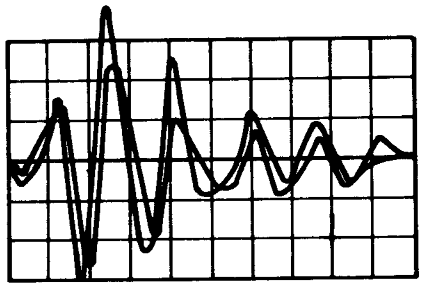
A. FUNCTION GENERATOR OUTPUT VOLTAGE
SWEEP: 80-MILLISEC
VERTICAL: 6-VOLTS FULL SCALE (1 VOLT/DIV)



B. SHAKER RESPONSE - ACCELERATION
SWEEP: 80-MILLISEC
VERTICAL: 3-VOLTS FULL SCALE (.5-VOLT/DIV)

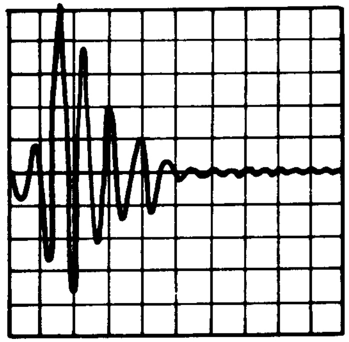


C. SIMULTANEOUS DISPLAY OF FUNCTION GENERATOR OUTPUT AND SHAKER RESPONSE
SWEEP: 80-MILLISEC
VERTICAL: FUNCTION GENERATOR OUTPUT. SAME AS A. SHAKER RESPONSE - SAME AS B

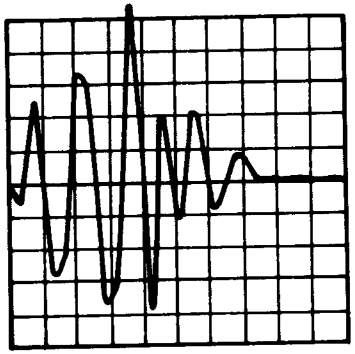


D. REPEAT SHOT WITH SAME CONDITIONS AS C TO ILLUSTRATE REPEATIBILITY OF DATA.

Fig. 7 - Bare table response characteristics of 5000-force-pound exciter to function No. 3 without input-signal compensation or shaker equalization

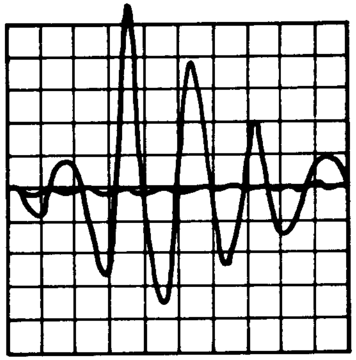


A. SWEEP: 250-MILLISEC (50-MILLISEC/DIV)
VERTICAL: 40-g FULL SCALE (4 g/DIV)



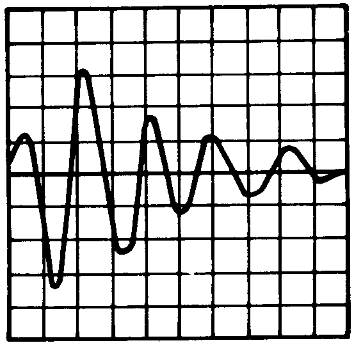
SWEEP: 90-MILLISEC
VERTICAL: 100-g FULL SCALE (10-g/DIV)
TABLE DISPLACEMENT: 11/32 INCH P-P

Fig. 8 - Response of 5000-force-pound exciter to function No. 1 with a 20-pound mass on the table and without compensation. NOTE: The above photograph was taken with the accelerometer mounted at the outer bolt ring.



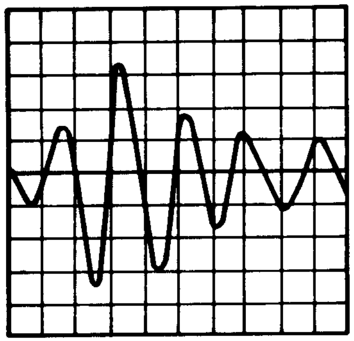
B. EXPANDED VIEW OF A
SWEEP: 20-MILLISEC/DIV
VERTICAL: 4-g/DIV

Fig. 9 - Response of 5000-force-pound exciter to function No. 2 with a 20-pound mass on the table and with compensation



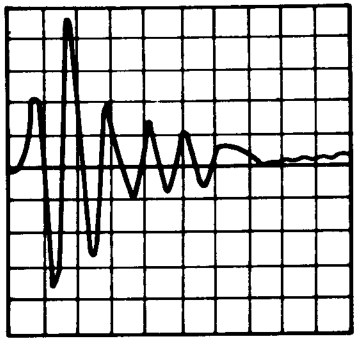
SWEEP: 25-MILLISEC (2-MILLISEC/DIV)
VERTICAL: 200-g FULL SCALE (20-g/DIV)

Fig. 10 - Response of 5000-force-pound exciter to function No. 2 with a 20-pound mass on the exciter and with compensation

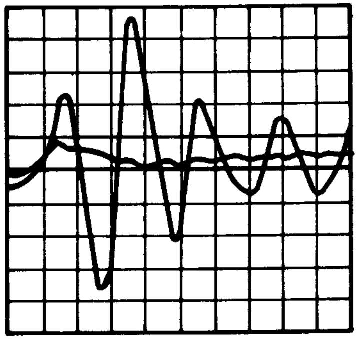


SWEEP: 25-MILLISEC/DIV)
VERTICAL: 40-g FULL SCALE (4-g/DIV)

Fig. 11 - Response of 5000-force-pound exciter to function No. 2 bare table and without compensation

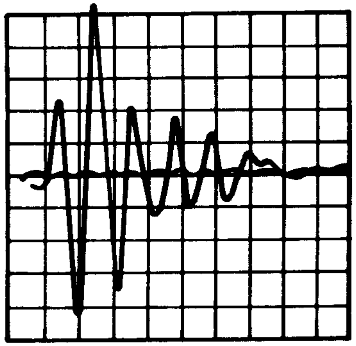


A. SWEEP: 80-MILLISEC (10-MILLISEC/DIV)
VERTICAL: 100-g FULL SCALE (10-g/DIV)

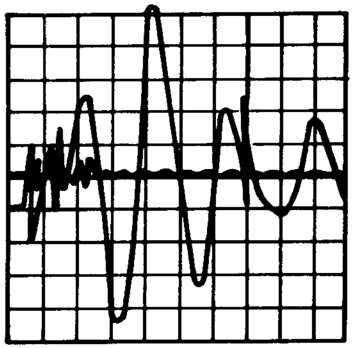


B. SWEEP: 80-MILLISEC (5-MILLISEC/DIV)
VERTICAL: 100-g FULL SCALE (10-g/DIV)

Fig. 12 - Response of 5000-force-pound exciter to function No. 3 (test No. 1) with a 20-pound mass on the exciter and without compensation

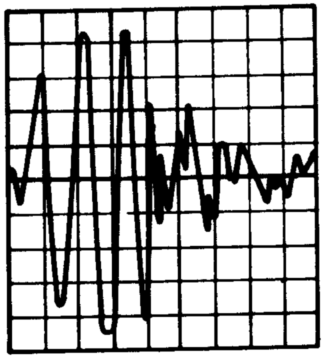


A. SWEEP: 10-MILLISEC/DIV
VERTICAL: 100-g FULL SCALE (10-g/DIV)



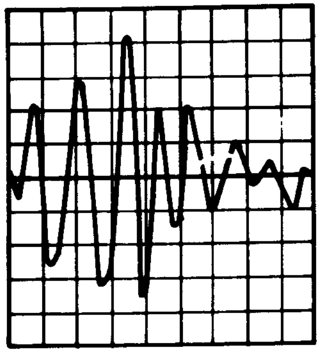
B. SWEEP: 5-MILLISEC/DIV
VERTICAL: 100-g FULL SCALE (10-g/DIV)

Fig. 13 - Response of 5000-force-pound exciter to function No. 3 (test No. 2) with a 20-pound mass on the exciter and without compensation



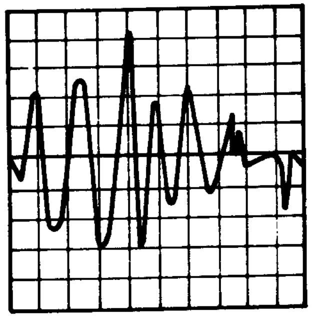
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV (40-g FULL SCALE)

Fig. 14 - Response of 5000-force-pound exciter to function No. 1 with a 41-pound test fixture and a 34-pound lead weight (75 pounds total) unbalanced and resonant exciter load without compensation

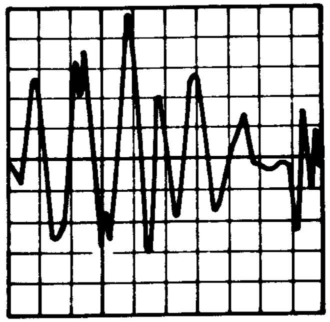


SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV (40-g FULL SCALE)

Fig. 15 - Response of 5000-force-pound exciter to function No. 1 (test No. 1) with a 41-pound test fixture without compensation

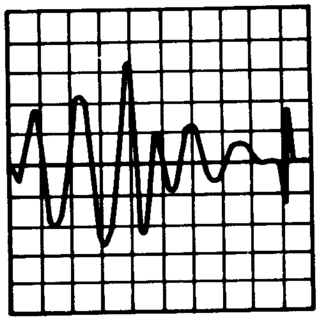


A. OUTPUT ACCELEROMETER NO. 1 (ON TABLE)
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV



B. OUTPUT OF ACCELEROMETER NO. 2 (ON FIXTURE)
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV

Fig. 16 - Response of 5000-force-pound exciter to function No. 1 (test No. 2) with a 41-pound test fixture without compensation



SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV

Fig. 17 - Response of 5000-force-pound exciter to function No. 1 (test No. 3) with a 41-pound test fixture without compensation

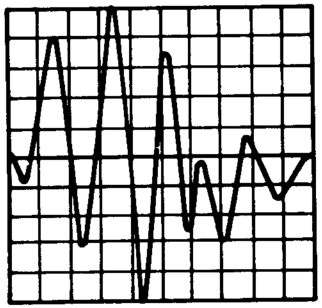
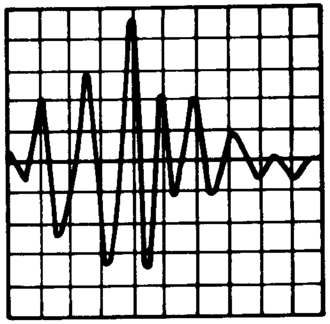
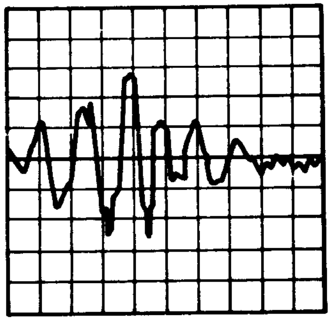


Fig. 18 - Electrical analog of function No. 1 for all function No. 1 results

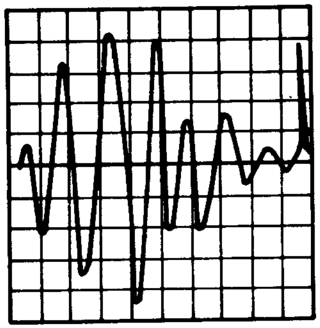


A. OUTPUT OF ACCELEROMETER NO. 1 (ON TABLE)
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV

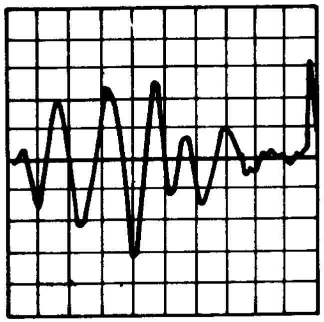


B. OUTPUT OF ACCELEROMETER NO. 2 (ON FIXTURE)
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV

Fig. 19 - Response of 5000-force-pound exciter to function No. 1 (test No. 1) with a 41-pound test fixture and with compensation

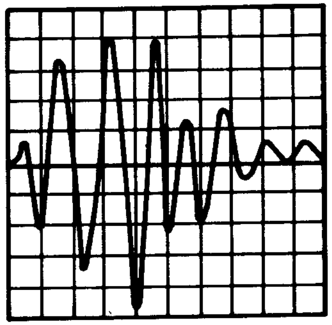


A. OUTPUT OF ACCELEROMETER NO. 1 (ON TABLE). SAME AS FIGURE 19 A
EXCEPT 180° OUT OF PHASE
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV

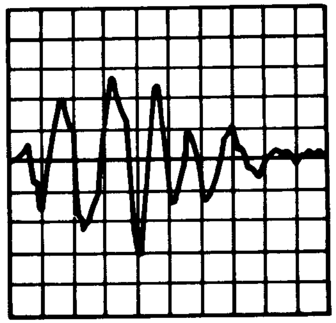


B. OUTPUT OF ACCELEROMETER NO. 2 (ON FIXTURE). SAME AS FIGURE 19 B
EXCEPT 180° OUT OF PHASE
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV

Fig. 20 - Response of 5000-force-pound exciter to function No. 1 (test No. 2) with a 41-pound test fixture and with compensation

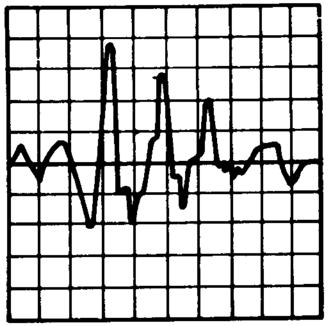


A. OUTPUT OF ACCELEROMETER NO. 1 (ON TABLE). SAME AS FIGURE 20 A
EXCEPT PHOTOCELL FLASHBACK ERROR AT THE END OF THE WAVEFORM
WAS ELIMINATED.
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV

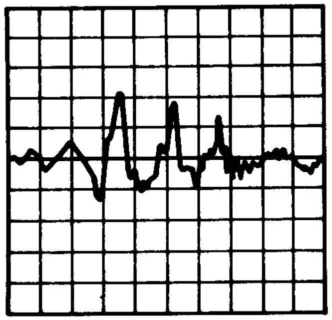


B. OUTPUT OF ACCELEROMETER NO. 2 (ON FIXTURE). SAME AS FIGURE 20 B
EXCEPT PHOTOCELL FLASHBACK ERROR AT THE END OF THE WAVEFORM
WAS ELIMINATED.
SWEEP: 10-MILLISEC/DIV
VERTICAL: 5-g/DIV

Fig. 21 - Response of 5000-force-pound exciter to function No. 1 (test No. 3) with a 41-pound test fixture and with compensation

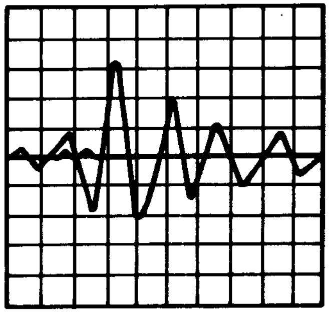


A. OUTPUT OF ACCELEROMETER NO. 1 (ON TABLE)
SWEEP: 200-MILLISEC (20-MILLISEC/DIV)
VERTICAL: 5-g/DIV

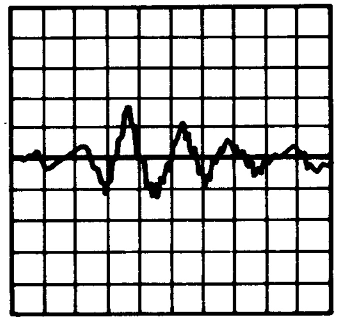


B. OUTPUT OF ACCELEROMETER NO. 2 (ON FIXTURE)
SWEEP: 200-MILLISEC (20-MILLISEC/DIV)
VERTICAL: 5-g/DIV

Fig. 22 - Response of 5000-force-pound exciter to function No. 2 with a 41-pound test fixture and without compensation

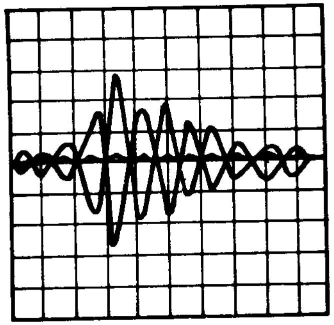


A. OUTPUT OF ACCELEROMETER NO. 1 (ON TABLE)
SWEEP: 200-MILLISEC (20-MILLISEC/DIV)
VERTICAL: 5-g/DIV

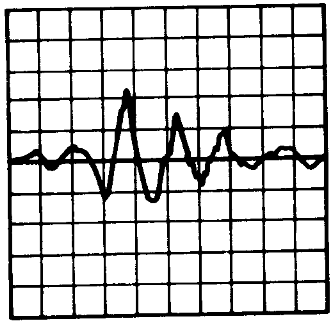


B. OUTPUT OF ACCELEROMETER NO. 2 (ON FIXTURE)
SWEEP: 200-MILLISEC (20-MILLISEC/DIV)
VERTICAL: 5-g/DIV

Fig. 23 - Response of 5000-force-pound exciter to function No. 2 (test No. 1) with a 41-pound test fixture and with compensation

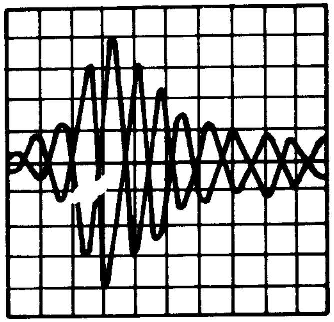


A. OUTPUT OF ACCELEROMETER NO. 1 (ON TABLE)
TWO SHOCK WAVEFORMS 180° OUT OF PHASE
SWEEP: 200-MILLISEC (20-MILLISEC/DIV)
VERTICAL: 5-g/DIV

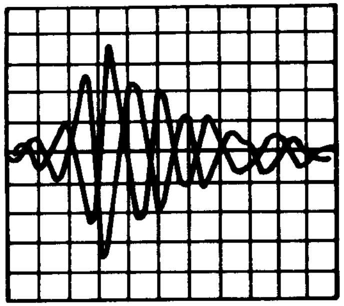


B. OUTPUT OF ACCELEROMETER NO. 2 (ON FIXTURE)
SWEEP: 200-MILLISEC (20-MILLISEC/DIV)
VERTICAL: 5-g/DIV

Fig. 24 - Response of 5000-force-pound exciter to function No. 2 (test No. 2) with a 41-pound test fixture and with compensation

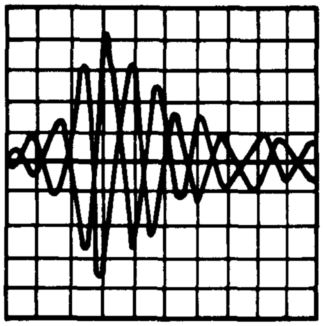


A. OUTPUT OF ACCELEROMETER NO. 1 (ON TABLE)
SWEEP: 50-MILLISEC (5-MILLISEC/DIV)
VERTICAL: 5-g/DIV

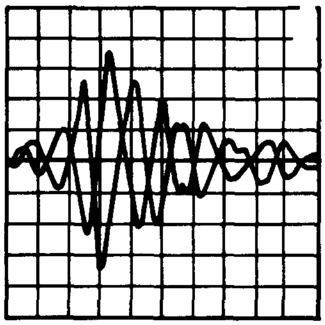


B. OUTPUT OF ACCELEROMETER NO. 2
SWEEP: 50-MILLISEC (5-MILLISEC/DIV)
VERTICAL: 5-g/DIV

Fig. 25 - Response of 5000-force-pound exciter to function No. 2 with a 41-pound test fixture and with compensation (two shock waveforms 180 degrees out of phase)



A. OUTPUT OF ACCELEROMETER NO. 1 (ON TABLE
SWEEP: 50-MILLISEC (5-MILLISEC/DIV)
VERTICAL: 5-g/DIV



B. OUTPUT OF ACCELEROMETER NO. 2 (ON FIXTURE)
SWEEP: 500-MILLISEC (5-MILLISEC/DIV)
VERTICAL: 5-g/DIV

Fig. 26 - Response of 5000-force-pound exciter to function No. 2 with a 41-pound test fixture and with compensation (two shock waveforms 180 degrees out of phase plus a reference trace)

As a matter of record, a Fourier analysis was performed on a function No. 2 transient wave shape to define its individual frequency contents. This analysis is shown in Figure 27.

The records (Figs. 5-26) were systematically arranged to demonstrate the degree of distortion and subsequently the effects of compensation and equalization. Even with increasing complexity of shaker loads with elastic masses, the simple compensating network was able to cope with the situation for the three given shock pulses. The similarity between the input and response functions is based on the compatibility of the two graphic forms, and, thus, firmly establishes the feasibility of this method of shock simulation. Records taken from tests conducted under equally controlled conditions indicate adequate repeatability to ensure proper control of qualification, reliability, and engineering evaluation test programs.

In the course of the development of the pulse function generator in the laboratory, attention was devoted to the features of adjusting and controlling the generator. Particular mention was made in Ref. 2 regarding difficulties encountered in adjusting and in maintaining adjustments through repeated application of the generator. It can be stated with assurance that none of these difficulties were encountered in a large number of tests. A further disadvantage of the use of the photocell function generator is mentioned in Ref. 2; that is, excessive shaker amplitude in the event of loss of control of the electron beam. With proper instrumentation, this event has a small margin of probable occurrence and, if desired, electronic safeguards can be taken against that.

The fact that the application of this type of shock simulation is not new, is pointed out in Refs. 1 and 2. What is new, however, is the introduction of a realistic shock pulse tailored to ground and flight test conditions. The uniqueness of this method, as compared with present positive shock pulse shape applications, is in the laboratory reproduction of the exact complex transient wave shapes that are typical of POLARIS ground and flight environment.

ADVANTAGES AND DISADVANTAGES

Advantages

- Environmental optimization of shock simulation; this avoids over- and under-testing with the customarily chosen positive shock pulses, particularly in the case of shock-mounted test items.
- The shaker system can be readily set up and checked out at low acceleration amplitudes, and the shock form can be repeated effectively at the 100-percent acceleration level.
- The simplification of directional shock testing.
- The selection of single test fixtures and test setups for shock and vibration simulations.
- Economy of shaker equipment utility and conservation of floor space otherwise devoted to singular purpose shock equipment.

Disadvantages

- Acceleration limitations at very low frequencies as a function of maximum shaker displacement. This limitation should not, however, become a detriment since the missile shock events generally have low acceleration amplitudes at the lower frequencies; and, further, a long stroke shaker could be designed.

- The high frequency limit is governed by the compensating network employed. It is anticipated, based on experience to date, that a 1500-cps frequency limit can be obtained with present commercial, specialized instrumentation.

CURRENT DEVELOPMENTS

Special note should be given to the shock pulse model to be used for the proposed shock simulation. A particular shock event, when simulated in the laboratory, should be founded on a statistically extreme environment, which can, under any actual condition, be exceeded only by a predetermined percentage. For obvious reasons, a single flight data set is not sufficient to describe the simulated shock environment. Mr. G. Warren Painter of Lockheed Aircraft Corporation, Burbank, California, is presently studying the problem of deriving a representative shock model by synthesis of several measured shock data samples of a single population. The success of this shock model investigation and electrodynamic shaker, shock-simulation-technique will enlarge the scope of present shock testing; in particular from two dimensions of the singular positive shock pulse (acceleration and frequency) to three dimensions of the arbitrary complex shock pulse (acceleration, frequency, and repeated number of acceleration peaks). The third dimension has a pronounced affect on the functional failures of the missile equipment.

Future Investigations

The method of simulation by arbitrary shock pulses in an electrodynamic shaker system has been proved feasible and thus further

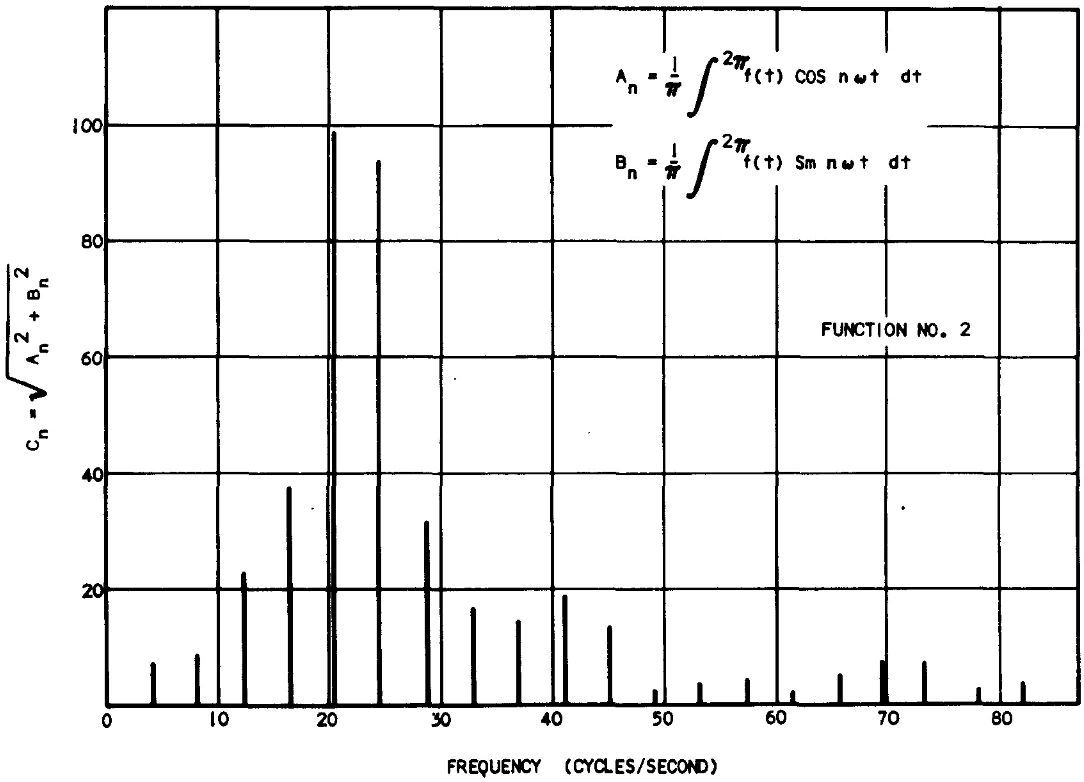


Fig. 27 - Plot of the Fourier transform data

investigation is warranted. Such an investigation will feature:

- The introduction of a commercially available shaker compensating and equalization network.

- Extension of the shaker system to 25,000 and 28,000 force-pounds.

- Improvement of recording techniques to afford an immediate spectral read-out of the transient model and shaker response for comparison of the two spectra.

- Extension of the test approach to include actual black box testing.

- Introduction of the synthesized shock environment model into the test program.

SUMMARY

It may be concluded that the feasibility of simulating three measured complex POLARIS shock pulses by electrodynamic shaker, has been demonstrated. Equally important is the realistic simulation of a shock environment which precludes excessive over-testing. Advantages are realized in the simplification of directional shock testing and in the selection of a single test fixture design for shock and vibration simulations. The introduction of the latest equalization network design is expected to extend the upper frequency band of the transient to approximately 1500 cps. The maximum acceleration amplitudes, which are limited by the shaker's displacement capabilities in the lower frequency bands (±0.5 inch for the 500-force-pound shaker), envelop, within a certain probability, the POLARIS shock environment. A study of measured shock data, to synthesize a realistic composite shock model, is presently in progress. This model will serve as the shaker input.

ACKNOWLEDGMENTS

Messers J. J. Labno, R. T. Kilduff, J. K. Owen, and others have provided invaluable assistance in the development of the photocell function generator and in conducting the supporting test program. The author also wishes to express his appreciation to Mr. J. E. Barkham for his valuable suggestions which greatly enhanced the scope of this project.

Appendix

PHOTOCELL FUNCTION GENERATOR

The photocell function generator consists principally of a compatible cathode-ray oscilloscope and photomultiplier tube. The cathode-ray tube selected for the investigation was a P11 blue phosphor with short persistence. The spectral range in angstrom units is 3770-5690 with a spectral peak of 4400 angstroms. The photocell selected was a 929 type blue sensitive with maximum response characteristics for wave lengths of 4000 angstroms and for a sensitivity of 0.042 microampere-per-microwatt.

The photocell, cathode follower, and control amplifier chassis are installed in a cylindrical container 6 inches in diameter and 14 inches long (Fig. 1). It is mounted to the oscilloscope in the manner shown in Fig. 2. An opaque mask, depicting in graphic form the required arbitrary shock function, is placed directly against the face of the oscilloscope and 8 inches away from the photocell.

For operation, apply filament and B+ voltage to the photocell circuit and connect the out-signal to the negative dc-input side of the oscilloscope plug-in differential preamplifier. The loop is closed and the function can be reproduced in electrical analog form with additional scope adjustments if necessary.

DISCUSSION

<u>Dr. Irwin, Chairman (NRL):</u> Mr. Schwabe, in obtaining one of these pulses which you wish the electrodynamic shaker to simulate, do you have to build the POLARIS and fly it and take a record from the flight in order to know what you want to simulate or how do you get this — who gives you this pulse which you are going to simulate with the electric shaker?

Mr. Schwabe: Admittedly, at first with a new missile generation being planned you have to extrapolate environment from previous missiles flown, but once the missile has been built and the first two or three flights have taken place, then we can synthesize the data as they come from the flight missile.

Mr. Booth (Autonetics): Is this high-fi technique you have for the shaker table, is that a closed loop system such that you can change the mass that you are shaking and still get repeatability from your shock pulses?

Mr. Schwabe: No, we use the open loop system as far as the entire shaker system goes.

Mr. Booth: Why not try a closed loop system?

Mr. Schwabe: We thought about it and we may go into it later on. We felt that the closed loop would involve more time and money, and therefore we would like to exploit the open loop approach, first.

Mr. Booth: Did you investigate the possibility of having the people who built the shaker table design this closed loop system so that you could get repeatability for any mass or system test?

Mr. Schwabe: We have not looked into it.

Mr. Sandler (Autonetics): This 180-degree phase changer that you employed to check the repeatability, when you tried to get an opposite polarity in your pulse, did you do this by merely reversing the voltage input?

Mr. Schwabe: Yes, just by throwing the switch and reversing the output of the function generator.

Mr. Galef (NESCO): Did I understand you to say that you can make all your adjustments at a low acceleration and then give it a shot at full acceleration and will work OK?

Mr. Schwabe: Yes, that is right. We attempt to make all the adjustments of compensation at a low acceleration level.

Mr. Galef: Well this sounds as if you are working only with linear systems then.

Mr. Schwabe: Admittedly, yes, I would say we have a supposition of linear system when we work at low-g levels.

Mr. Stewart (Douglas): How does your compensation network differ from the usual peak notch filters used in random vibration testing?

Mr. Schwabe: All that we used are some compacitance and resistance networks. Just very simple networks because the frequencies of these shock models were not very high and for relatively low frequencies, up to about 400-500 cycles, we could get away with the simple networks.

Mr. Stewart: Do you intend in the future to use the peak notch filters?

Mr. Schwabe: Yes, that is, a commercially available unit.

Mr. Stewart: Do you have any idea of what would happen if you tried to use the band pass equalization network?

Mr. Schwabe: No, I'm not a test engineer as such, I'm an environmental engineer and I could not answer the question from a strictly instrumentation standpoint.

Mr. Cohen (Sylvania): How would this method be used in a simulation of a ramp input shock?

Mr. Schwabe: I would say, off hand, there is no limitation there. I, may be talking off the top of my head but giving you an answer at short notice, would say that it could be quite easily accomplished with this setup. You may have to go to a different function generator, but I believe there are certain things in the wind right now with function generators with which you probably could do it very easily.

Mr. Cohen: Would it be possible to dampen out the remainder of the shock pulses that you're getting in this thing, so that you would get perhaps one clean shock and that's all?

Mr. Schwabe: As you notice, these three models I showed, are tapered off. We customarily use a half sine ending up with a very high velocity which we did not encounter with our shock models, therefore we were not in any difficulty with limitation of shaker displacement or shaker travel.

Mr. Nankey (GE): What was the resonant frequency of the fixture that was used in these experiments — the dominant mode?

Mr. Schwabe: As far as I know, the shaker frequencies were relatively high. When we had

the third configuration on the shaker table, the 41-lb mass, we intentionally put some cantilever beams on there and we got a certain amount of feedback into the system, but we were able to compensate for it.

Mr. Kirby (Motorola): Did you find it necessary in any way to alter the amplifier response characteristics, such as feedback networks?

Mr. Schwabe: Of the function generator?

Mr. Kirby: No, of the power amplifier.

Mr. Schwabe: No, none whatsoever. All that we needed was to check out and calibrate the particular power amplifier and check out the frequency response, so long as the frequency response is fairly flat and the power tubes are balanced we don't have any problem.

Mr. Kirby: Did you have any problem with armature resonance?

Mr. Schwabe: Yes, we had armature resonance, that's the reason we had to compensate the armature resonance as well as the additional weight which changes the compacitance of the network.

* * *

RELATIONSHIPS BETWEEN RANDOM VIBRATION TESTS AND THE FIELD ENVIRONMENT

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On the hypothesis that vibration tests of greatly differing character can only be compared in terms of the responses of elements of typical devices to these tests, computations are made for these responses and for the desired responses to ideal tests. For band equalized random vibration systems, the computed responses are compared to the desired responses for various specimen to vibration exciter mass ratios, specimen damping factors, and frequency to control bandwidth ratios. For sweep random systems, the achievable accuracy is discussed and shown to be of the same order as accuracies achievable by band equalized systems. A hybrid system, incorporating a few peak-notch equalizers in addition to the multiband control, overcomes the deficiencies of band equalized systems. A warning: the reliability of equipment tested by band equalization systems having large control bandwidths is questionable.

VIBRATION TEST SYSTEM

Vibration test systems may be characterized by the system illustrated in Fig. 1. Such systems have a vibration generator with output to input transfer function $H_{1(f)}$ and a control with gain A, and are used to test devices or specimens having elements with transfer functions $H_{2(f)}$. Since vibratory level a_t to be applied to the specimen, this level is monitored; manual or automatic means are used to adjust the gain A to maintain that desired level.

Although gross comparisons of vibration tests of the same type may be made frequently in terms of the vibratory level a_t applied to the specimen, the only precise way to compare tests of differing character is by comparing the responses a_s of elements of typical specimens for various tests to the response desired from an ideal test.

The schematic of Fig. 2 shows this vibration test system, simplified to include only those elements necessary to compare vibration tests of differing types.[1] An element in the

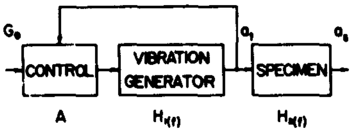

Fig. 1 - Generalized vibration test system showing means to control vibratory level to test specification

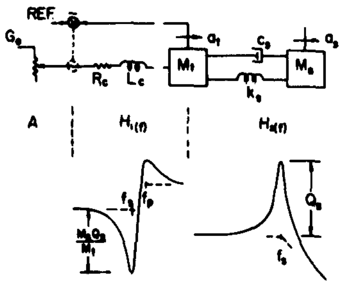

Fig. 2 - Schematic at vibration test system, simplified to retain only those elements necessary to compare vibration tests of differing types

[1] Harris, C.M., and Crede, C. E., Shock and Vibration Handbook (McGraw-Hill Book Company, Inc., New York), Vol. 2, Chap. 25, p. 31.

specimen may be defined in terms of its mass M_s, spring constant k_s, and damping c_s. Important parameters of the vibration exciter are the moving table mass M_t, and the resistance R_c and inductance L_c of the driving coil. For each control band of a band equalized random vibration test system[2] and for a sweep random vibration test system[3] the gain A of the control is frequency independent and is caused to vary in amplitude to keep the table acceleration a_t equal to a desired reference value (REF.). Although the elements to the left of the gap in the schematic are electrical in nature, a transformation to equivalent mechanical units simplifies computations without loss of accuracy. The input G_o may then be expressed in terms of mean square acceleration spectral density, commonly called acceleration density or power spectral density, in units of g^2/cps. The table acceleration a_t and specimen element mass acceleration a_s then are in terms of the gravitational acceleration g.

The transfer function a_s/a_t of the element of the specimen is defined as

$$H_{2(f)} = \frac{a_s}{a_t} = \frac{1}{1 - \left(\frac{f}{f_s}\right)^2 + \frac{j}{Q_s}\left(\frac{f}{f_s}\right)}, \quad (1)$$

where f_s is the undamped natural frequency of the element of the specimen mounted on an infinite mass, and Q_s, the magnification factor, is the reciprocal of twice the ratio of the actual damping to the critical damping. The shape of the response is illustrated immediately below the specimen components.

The transfer function of the vibration generator is affected by both, the parameters of the vibration generator and by the specimen. Although even for this simplified system the complete expression for the transfer function is quite complex,[4] the expression may be split into two factors of which one is related solely to the parameters of the vibration generator, and the other includes the effect of the specimen resonance. The term which relates solely to the parameters of the vibration exciter varies only gradually with frequency. In contrast, however, the magnitude of the term which includes the effect of the specimen resonance varies much more rapidly with frequency, usually orders of magnitude more rapidly.

Since any vibration test which can cope effectively with this second term can adequately handle the first term, a simplifying reduction is made; the transfer function of the vibration exciter is assumed to consist only of the rapidly varying second term, defined as $H_{1(f)}$, which has the form

$$H_{1(f)} = \frac{1 - \left(\frac{f}{f_s}\right)^2 + \frac{j}{Q_s}\left(\frac{f}{f_s}\right)}{1 - \left(\frac{f}{f_p}\right)^2 + \frac{j}{Q_p}\left(\frac{f}{f_p}\right)}, \quad (2)$$

and the notch-peak shape shown in Fig. 2. f_p and Q_p are the resonant frequency and magnification factors for the peak, which is usually more highly damped than the notch due to currents flowing in the driver coil resistance. The depth of the notch is approximately $M_s Q_s/M_t$ as shown. This notch-peak or peak-notch fluctuation is commonly observed in the measurement of the transfer function of actual systems.[5]

The ratio of the peak frequency to the specimen resonant frequency is given by[1]

$$\left(\frac{f_p}{f_s}\right)^2 = \frac{M_t + M_s}{M_t} . \quad (3)$$

RESPONSE OF BAND EQUALIZED SYSTEMS

In a band equalized system,[2] the test spectrum is divided into a number of adjacent frequency bands, each of which has a control element with gain A. At low frequencies, it is common that the width B of the frequency band is much wider than the bandwidth f_s/Q_s of the specimen resonance, and also wider than the peak to notch frequency spacing $f_p - f_s$. Conversely, at high frequencies, the width B of the frequency band is commonly narrower than the bandwidth of the specimen resonances.

The level of vibration a_t applied to the specimen is adjusted for each band until

$$a_t^2 = B G_o , \quad (4)$$

where B is the control bandwidth in cps and G_o is the desired acceleration density. (The same

[2]Maki, C. E., "IRE Transactions on Audio," Vol. AU-8, No. 6 (Nov.-Dec. 1960).
[3]Booth, G. B., "Sweep Random Vibration," Institute of Environmental Sciences (April 1960).
[4]Unholtz, K., "The Influence of Electrical and Motional Impedance on the Control and Performance of Some Vibration Machines," ASME (June 1956).

[5]Booth, G. B., "Random Motion Test Techniques," Institute of Environmental Engineers (April 1959).

value is used as the input to the system without loss of generality since A has not been specified.)

For an ideal test, the spectrum of the acceleration a_t applied to the specimen is flat with the value G_o. For this ideal test, the desired response of the mass M_s is defined to have the rms value a_o, where

$$a_o^2 = G_o \int_0^\infty H_{2(f)}^2 \, df = \frac{\pi}{2} f_s Q_s G_o . \tag{5}$$

For an actual band equalized test, the response acceleration a_s of the specimen differs from the desired value since the spectrum of a_t is not flat. The magnitude of this difference may be determined by calculating a_s for various bandwidths B, frequencies f_s, and magnification factors Q_s.

For each control band m, the rms response a_{tn} for that band may be calculated from

$$a_{tn}^2 = A_n^2 G_o \int_{f_n - B/2}^{f_n + B/2} H_{1(f)}^2 \, df . \tag{6}$$

However, the control system adjusts the gain A, either manually or automatically, until

$$a_{tn}^2 = G_o B , \tag{7}$$

as is required by the test specification, giving

$$A_n^2 = \frac{B}{\int_{f_n - B/2}^{f_n + B/2} H_{1(f)}^2 \, df} . \tag{8}$$

The acceleration response of the specimen mass may now be calculated as the sum of the responses for each of the individual bands

$$a_s^2 = G_o \sum_n A_n^2 \int_{f_n - B/2}^{f_n + B/2} H_{1(f)}^2 H_{2(f)}^2 \, df , \tag{9}$$

but since A_n is adjusted in accordance with Eq. (8), a_s becomes

$$a_s^2 = G_o B \sum_n \frac{\int_{f_n - B/2}^{f_n + B/2} H_{1(f)}^2 H_{2(f)}^2 \, df}{\int_{f_n - B/2}^{f_n + B/2} H_{1(f)}^2 \, df} . \tag{10}$$

Since the desired value of the response a_s is given by Eq. (5), a convenient comparison of the actual response to the desired response is given by

$$\frac{a_s^2}{a_o^2} = \frac{2B}{\pi f_s Q_s} \sum_n \frac{\int_{f_n - B/2}^{f_n + B/2} H_{1(f)}^2 H_{2(f)}^2 \, df}{\int_{f_n - B/2}^{f_n + B/2} H_{1(f)}^2 \, df} . \tag{11}$$

Since H_1 and H_2 are the relatively complex functions given by Eqs. (1) and (2), evaluation of Eq. (11) is difficult. By various techniques, however, approximate solutions give the results summarized in Figs. 3, 4, and 5.

In Figs. 3, 4, and 5, the mass M_t includes the mass of the moving element and all portions of the specimen mounting fixture and specimen

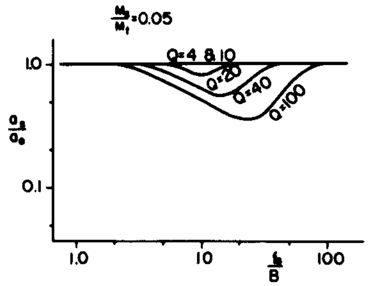

Fig. 3 - Band equalized random vibration test: ratio of calculated response to desired response for resonating mass equal to 0.05 times the table mass

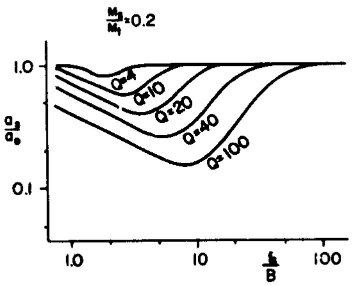

Fig. 4 - Band equalized random vibration test: ratio of calculated response to desired response for resonating mass equal to 0.2 times the table mass

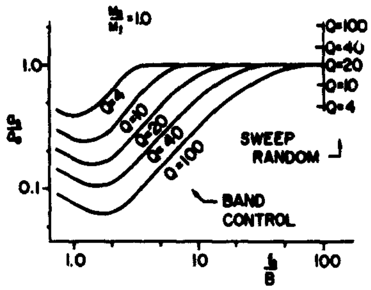

Fig. 5 - Band equalized random vibration test: ratio of calculated response to desired response for resonating mass equal to the table mass. Sweep random test: ratio of calculated response to desired response shown at right.

not participating in the resonant response. Since the desired response is a_o, the deviation of the calculated value for a_s/a_o from 1.0 is the error. The abscissa is plotted in generalized form, as the ratio of the resonant frequency f_s of the specimen element to the control bandwidth B.

In all three figures the maximum error occurs in the frequency region where the difference between the peak frequency and the specimen resonant frequency is of the same magnitude as the bandwidth B.

As an example, consider control bandwidths with width $B = 25$ cps. The response of a small resonating device with mass of 0.05 times the table mass and $Q = 40$, may be expected to be only 60 percent of the desired response if the resonant frequency is near 350 cps, but would be nearly correct if the resonant frequency were 800 cps.

As the curves of Figs. 4 and 5 show, larger resonating masses cause increasing errors at lower frequencies. These errors are particularly serious since the specimen is inadequately tested. Note that for large control bandwidths, for example 100 cps, and large resonating masses, $M_s/M_t = 1$ (see Fig. 5), objects with Q's of 40 are tested at less than one-half the desired level at all frequencies below 900 cps, and if the Q's are 100, at all frequencies below 1700 cps.

SWEEP RANDOM RESPONSE

The sweep random test provides an intense narrow-band random vibration excitation to the specimen, sweeping slowly over the frequency range.[3] The excitation level and sweep rate are so chosen that at every level of response of a_s the number of acceleration peaks for the sweep random test is identical to the number of acceleration peaks in the ideal test described with reference to Eq. (5). Although no assumptions are necessary regarding the type of damping in the specimen, linearity with drive level, or shape of the S-N curve, in order to achieve the desired reduction in the cost of test equipment, it is necessary to test the various resonances sequentially using a single control channel to adjust the gain A rather than the large number of parallel channels used in band equalized systems.

In a sweep random test, the rms excitation level, a_t, is maintained at the level σ_e derived in Ref. 3:

$$\sigma_e = q\sqrt{\frac{\pi G_o f_s}{2Q_o}}, \tag{12}$$

where G is a constant related to the desired test time and a reference magnification factor. Although a value of 10 for Q_o was previously recommended in Ref. 3, the results of this paper show that a value for $Q_o = 20$ results in lower errors for the expected range of specimen magnification factors Q_s.

Using this new recommended value for Q_o, Eq. (12) becomes

$$\sigma_e = q\sqrt{\frac{\pi G_o f_s}{40}}. \tag{13}$$

Even though the control system maintains a_t equal to σ_e, the rms response a_s will exceed the desired value for specimen magnification factors Q_s which exceed 20, and will be less than the desired value for Q_s less than 20. The error is within ±3 db for a range in Q_s from 10 to 40, and within ±7 db for a range in Q_s from 4 to 100. This error is plotted at the right of Fig. 5 to the same scale as the band equalized curves at the left.

In addition to the error just described, the sweep random test is subject to the same errors as the band equalized test, and the curves to the left of Figs. 3, 4, and 5 apply. However, these inaccuracies are much less serious for the sweep random test than for the band equalized system, since the control bandwidth is normally much smaller, 3 cps, and tests are normally conducted only on the right side of the diagrams, for values of f/B greater than 7 corresponding to frequencies above 20 cps. In addition, the errors characteristic of the sweep random test (shown at the right in Fig. 5) correct, to a very

large degree, the errors due to a finite bandwidth (shown at the left in Fig. 5).

In general, the errors of the sweep random test are of the same order as the errors of the band equalized test, and may be less or greater depending upon the frequency of resonance and the size of the resonating mass. The sweep random test is particularly suitable for testing large objects, those for which M_s/M_t is one or more, and, of course, for all objects when equipment cost is a major factor.

PEAK NOTCH AND HYBRID SYSTEMS

In a peak-notch equalized system, the control element of Figs. 1 and 2, having frequency independent gain A, is replaced by an equalizer having a transfer function the inverse of $H_{1(f)}$. Theoretically, such a system accurately corrects for resonances with any magnification factor Q_s or any ratio of M_s to M_t, and provides appreciably lower errors than either the band equalized system or the sweep random system previously described.

The peak-notch system has two deficiencies: 1) correction for change in damping within the specimen is difficult to make during a test; and 2) adjustment of equalizers is so time consuming that, rather than reduce the percentage of total equipment time used for performing tests, accuracy is sacrificed.

It is this second deficiency which precipitated the development of the automatic band equalized system, the multiband system, and the subsequent decline in popularity of the peak-notch systems.

A hybrid system, which uses a few peak-notch equalizers in addition to a complete band equalized system, offers attractive possibilities. For many of the specimens now being tested, most of the resonances are in the higher frequency ranges where the band equalized systems perform well, however, a few resonances in the specimen or fixture occasionally occur at low frequencies. A brief study of the unequalized response, a_t, of the vibrator table with specimen attached, and reference to the curves of Figs. 3, 4, and 5, will show the resonances for which peak-notch equalizer compensation is necessary. Peak-notch equalizer adjustment at these low frequencies can be easily and rapidly done.

Appendix

EVALUATION OF INTEGRALS

LOW-FREQUENCY REGION: $B > f_p - f_s$

To get useful numbers for Eq. 11, the integrals in both the numerator and the denominator must be evaluated. The low-frequency numerator integral, I_{NL}, is

$$I_{NL} = \int_{f_n - B/2}^{f_n + B/2} H^2_{1(f)}\, H^2_{2(f)}\, df$$

$$= \int_{f_n - B/2}^{f_n + B/2} \frac{df}{\left[1 - \left(\frac{f}{f_p}\right)^2 + \frac{j}{Q_p}\left(\frac{f}{f_p}\right)\right]^2} . \qquad (14)$$

Since most of the contribution to this integral is in the frequency region near f_p, the integral over the frequencies from $f_n - B/2$ to $f_n + B/2$, including f_p, is approximately the same as the integral from 0 to ∞, which is known to have the value

$$I_{NL} = \frac{\pi}{2} f_p Q_p . \qquad (15)$$

The low-frequency denominator integral, I_{DL}, is

$$I_{DL} = \int_{f_n - B/2}^{f_n + B/2} H^2_{1(f)}\, df$$

$$= \int_{f_n - B/2}^{f_n + B/2} \left[\frac{1 - \left(\frac{f}{f_s}\right)^2 + \frac{j}{Q_s}\left(\frac{f}{f_s}\right)}{1 - \left(\frac{f}{f_p}\right)^2 + \frac{j}{Q_p}\left(\frac{f}{f_p}\right)}\right]^2 dF. \qquad (16)$$

The major contribution to this integral is also in the frequency region near f_p. Since the squared height of the peak is approximately

$$\left(\frac{M_s}{M_t} Q_p\right)^2$$

and the width of the peak is f_p/Q_p, a parameter c^2 may be defined such that

$$I_{DL} = c^2 f_p Q_p \left(\frac{M_s}{M_t}\right)^2 . \qquad (17)$$

The imaginary term in the numerator may be neglected and Eq. (16) graphically integrated for various mass ratios and Q_p. The resulting approximate values for c are shown in Fig. 6.

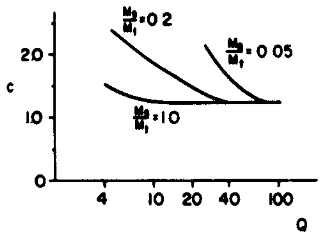

Fig. 6 - Approximate values of c

Equation (11), for the ratio of the actual response to the desired response in the low-frequency region, by the use of Eqs. (15) and (17), becomes

$$\frac{a_s^2}{a_o^2} = \frac{B}{f_s Q_s c^2}\left(\frac{M_t}{M_s}\right)^2 , \tag{18}$$

and defining a control bandwidth to resonance bandwidth factor γ,

$$B = \gamma \frac{f_s}{Q_s} . \tag{19}$$

Equation (11) becomes

$$\frac{a_s}{a_o} = \frac{\sqrt{\gamma}}{c} \times \frac{M_t}{Q_s M_s} \qquad \text{(Low-frequency region)} \tag{20}$$

which was used to calculate the left portion of Figs. 3, 4, and 5.

HIGH-FREQUENCY REGION

$B < f_p - f_s$ AND $B > 4\frac{f_s}{Q_s}$

In this high-frequency region, the excitation of the specimen occurs primarily in the frequency region around the specimen resonant frequency f_s.

The high frequency denominator integral, I_{DH}, is

$$I_{DH} = \int_{f_n - B/2}^{f_n + B/2} H^2_{1(f)}\, df$$

$$= \int_{f_s(1-a/Q)}^{f_s(1+e/Q)} \left[\frac{1 - \left(\frac{f}{f_s}\right)^2 + \frac{j}{Q_s}\left(\frac{f}{f_s}\right)}{1 - \frac{M_t}{M_s + M_t}\left(\frac{f}{f_s}\right)^2 + \frac{j}{Q_p}\left(\frac{f}{f_p}\right)}\right]^2 df, \tag{21}$$

where a and e are chosen to place the control band B on the notch of $H_{1(f)}$ (Fig. 2). Defining a notch depth parameter h as

$$h = \frac{M_t}{Q_s M_s} , \tag{22}$$

where $h < 1$, and neglecting small terms, I_{DH} becomes

$$I_{DH} = \left(\frac{M_s + M_t}{Q_s M_s}\right)^2 \times \frac{f_s}{Q_s h} \times \left\{ \begin{aligned} & 2(e^2 - a^2) + \frac{4a(a+e)}{1-ah} - \frac{2(a+e)^2}{(1-ah)^2} + \frac{4}{3}\frac{h(a+e)^3}{(1-ah)^3} \\ & - \frac{h^2(a+e)^4}{(1-ah)^4} + \frac{4}{5}\frac{h^3(a+e)^5}{(1-ah)^5} - \frac{a+e}{(1-ah)(1+eh)} \\ & + 4j\left[\frac{eh}{(1-ah)(1+eh)} - \frac{h(a+e)^2}{(1-ah)^2}\right] \end{aligned} \right\} . \tag{23}$$

A parameter p is defined such that

$$I_{DH} = \gamma^2 p f_s \left(\frac{M_s}{M_t}\right)\left(\frac{M_s + M_t}{Q_s M_s}\right)^2 , \tag{24}$$

and Eq. (23) evaluated for $e = 1.5$, $a = 0$, 3.5, 8.5, and 18.5 (values of γ equal to 1.5, 5, 10, and 20) for various mass ratios and Q_s. The resulting values for p are shown in Fig. 7.

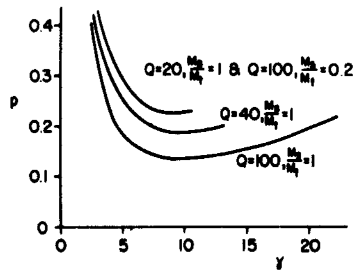

Fig. 7 - Value of p

The high-frequency numerator integral, I_{NH}, is

$$I_{NH} = \int_{f_s - B/2}^{f_s + B/2} H^2_{1(f)} \, H^2_{2(f)} \, df$$

$$= \int_{f_s(1-a/Q)}^{f_s(1+e/Q)} \frac{df}{\left[1 - \frac{M_t}{M_s + M_t}\left(\frac{f}{f_s}\right)^2 + \frac{j}{Q_p}\left(\frac{f}{f_p}\right)\right]^2} . \tag{25}$$

Since the imaginary term contributes little to this integral, taken well down on the low-frequency slope of the $H_{1(f)} H_{2(f)}$ curve, the imaginary term may be omitted and I_{NH} integrated to give

$$I_{NH} = \frac{1}{2}\left[\frac{f_s\left(1 + \frac{e}{Q}\right)}{1 - \frac{M_t}{M_s + M_t}\left(1 + \frac{e}{Q}\right)^2} - \frac{f_s\left(1 - \frac{a}{Q}\right)}{1 - \frac{M_t}{M_s + M_t}\left(1 - \frac{a}{Q}\right)^2}\right.$$

$$+ \frac{f_s}{4}\left(\frac{M_s + M_t}{M_t}\right)^{1/2} \log_e \left[\frac{1 + \left(1 + \frac{e}{Q}\right)\left(\frac{M_t}{M_s + M_t}\right)^{1/2}}{1 + \left(1 - \frac{a}{Q}\right)\left(\frac{M_t}{M_s + M_t}\right)^{1/2}}\right.$$

$$\left.\left. \times \frac{1 - \left(1 - \frac{a}{Q}\right)\left(\frac{M_t}{M_s + M_t}\right)^{1/2}}{1 - \left(1 + \frac{e}{Q}\right)\left(\frac{M_t}{M_s + M_t}\right)^{1/2}}\right]\right] . \tag{26}$$

A parameter K_1, is defined such that

$$I_{NH} = K_1 \gamma \frac{f_s}{Q_s}\left(\frac{M_s + M_t}{M_s}\right)^2 , \tag{27}$$

and Eq. (26) evaluated for $e = 1.5$ and $a = 3.5$, 8.5, and 18.5 ($\gamma = a + e = 5, 10, 20$) for various mass ratios and Q_s. Values for K_1 are shown in Fig. 8.

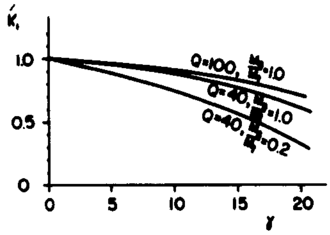

Fig. 8 - Values of K_1

Equation (11), for the ratio of the actual response to the desired response in the high-frequency region, using Eqs. (24) and (27) becomes

$$\frac{a_s^2}{a_o^2} = \frac{2}{\pi} \times \frac{K_1}{pQ_s} \times \frac{M_t}{M_s} , \quad \text{(High-frequency region)} \tag{28}$$

which was used to calculate high Q points to the right of Figs. 3, 4, and 5.

NOTE

In neither the low-frequency region nor the high-frequency region does the peak magnification factor Q_p greatly affect the ratio of the actual response to desired response. This is convenient, since Q_p is strongly affected by damping parameters of the specific vibration generator being used.

DISCUSSION

M. Oleson (NRL): Galt if I understand your derivation right, this is all for an uncompensated table. Is that correct?

Mr. Booth: Yes, it's an uncompensated table. Remember we are talking about band equalization and this is many parallel bands. My derivation was for any one band of the total so there is no compensation within a band — no peak notch equalizers; we're using the adjustment of the levels of the individual bands to do our compensation.

Mr. Oleson: So your remarks pertaining to the narrow-band swept test for example, again if I understand you right, wouldn't incorporate the effects of feedback control of the table gain, is that correct?

Mr. Booth: No, they do incorporate the feedback control of the table gain because in a similar way the level at the table is used to control the gain variable A in Fig. 1. Figure 1 applies for the sweep random vibration as well as the band vibration. In general, the sweep

random vibration has one band and in the more recent systems there is automatic control of the level of this one band. In the most recent multiple band systems there are many bands side by side and automatic control of the level of each of these multiple bands.

Dr. Vigness (NRL): Mr. Booth has shown very nicely that we are undertesting but from the viewpoint that it might be detrimental. He is illustrating something that we have been trying to show for a long while. I would like to go back a little bit and not begin with the specifications as given but consider how the specifications were derived. The specifications are generally derived by obtaining a large group of field measurements and then taking an envelope of these field measurements. When you have taken the field measurements you then are concerned primarily with the maximum values. If you have to go out in the field and make measurements and observe the natural frequencies of the equipment in their locations, take the measurements with the equipment there, you would find at these so-called fixed-base natural frequencies that you would have very low values of vibrations under the field conditions. These are not taken into account in the specifications at all. If we were to permit this decrease in excitation at these particular frequencies then we would be doing automatically what we would like to be doing if we were going to simulate the real field conditions. To me, if you have a relatively narrow bandwidth, such as is presently in use, and you take the average level of that bandwidth to be correct, then permit the equipment to react back on the machine, assuming that the vibration table itself has at least as great an impedance as the foundation of the equipment in the field, then we would not be under-testing. We would be more realistic. As it is, the way I look at it, if you follow the specification to the letter, you are over-testing and you will break up your equipment and your machine. So I would much prefer not to make this correction, but to allow the machine, provided its impedance is as great as the foundation in the field, to take its own average level over this relatively narrow bandwidth and let that be the solution to the problem. To me, to try to get an exactness of a certain level when a very large reaction of a piece of equipment on the shock machine or the vibration machine (it works for both) is not realistic. We shouldn't do that. I think it is very nice of you to bring the problem out, but I'm on the other side of the fence.

Mr. Booth: Well, in answer to this I would like to comment that the problem is one of making the impedance of the machine like the impedance of the field mounting point. If we could do that and be assured that we have it then I would certainly agree with Dr. Vigness but in some frequency ranges the impedance of the vibration machine is not high which, unfortunately complicates the problem.

* * *

MULTI-PLANE VIBRATION TESTING TECHNIQUES

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Three methods of combining three planes of vibration into a single test are discussed. Vibration monitor locations, instrumentation, and presentation of data required for multi-plane vibration tests will be described. It is shown that more data are required before this technique can become a universal tool.

INTRODUCTION

In this decade, much attention has been focused on the development of laboratory environmental tests to simulate the environments encountered in actual service. These laboratory environmental tests permit economical evaluations to be made of operational systems prior to actual flight. However, the customary three-plane vibration tests can produce fallacious test results, especially when vibration is combined with other environments.

This paper discusses considerations for determining laboratory test requirements from pre-flight and flight data and three methods of combining three planes of vibration into a single test to detect anomalies which would occur during service.

TEST REQUIREMENT CONSIDERATIONS

Duplication of the service anomalies requires a thorough analysis of the mission vibration characteristics and their effect on vehicle hardware as modified by hardware mounting structure and other environments. Vibration testing to arbitrary specifications can cause unrealistic failures which require redesign at the cost of time, money, and usually, weight. To minimize over-testing the following elements should be considered in preparing a vibration specification:

- The total time various vibration stress levels are expected during the mission.
- The modification of these vibration stress levels produced by the vehicle mounting structure.
- The effect of other environments which will modify the vibration stresses.
- The point on the specimen for which the responses must be defined.
- The presentation of the minimum laboratory vibration data which assures compliance with the requirements.

Supplementing pre-flight and flight vibration information with laboratory transmissibility data enables laboratory vibration test requirements to be determined with a considerable degree of accuracy. Such transmissibility data are of help in defining the stress level modification produced by the vehicle structure.

A complete missile undergoing a laboratory transmissibility test is shown in Fig. 1. It should be noted that vibrators are attached to the vehicle at the points where the majority of the vibration originates and are oriented off-axis to obtain responses in the horizontal, vertical, and thrust axes simultaneously.

CONVENTIONAL THREE-PLANE TESTING SHORTCOMINGS

Vibration test requirements are usually referred to the three principal planes of the test specimen. However, test requirements permitting individual plane testing in each of these three separate planes cannot provide the actual motion requirements for several reasons. Some of the main reasons are:

- The vibration test time does not consider the effect of the extraneous side motion.
- The interaction of the off-axis vibration cannot be detected.

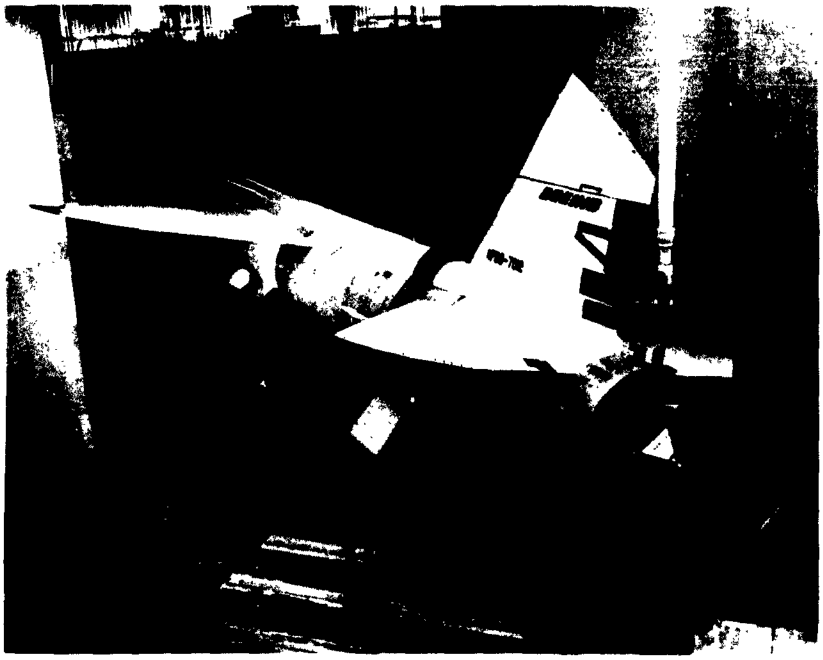

Fig. 1 - Test setup for obtaining complete vehicle transmissibility data

• The total effects of the vehicle mounting structure are not evaluated.

The multi-plane testing technique proposed in this paper will eliminate several of these shortcomings.

SINGLE OBLIQUE-PLANE TESTING

The relationship between several vibration planes of a specimen can be illustrated by considering a cube which represents any point on a specimen. Figure 2 shows accelerometers (X, Y, Z) located in the three principal planes and additional accelerometers (X', Y', Z') located in three other mutually perpendicular planes with one of these planes (X') being perpendicular to the diagonal of the cube. Vibrating this cube at ±10 g in each of the principal planes and at ±17 g along the diagonal of the cube produces acceleration levels in the six planes as shown in Table 1.

The acceleration levels in the principal planes can be duplicated by a single test along the diagonal of the cube, although it is necessary to increase the exciting force by a factor of 1.7 to obtain these levels. Vibration in each of the three principal planes simultaneously can produce acceleration levels identical to those obtained by a diagonal-of-the-cube test. Since the vibration levels in the X', Y', and Z' planes are usually not specified for the normal three-plane test, it does not necessarily follow that vibration along the diagonal of the cube is an over-test.

Fig. 2 - "Cube" with two sets of three mutually perpendicular accelerometers

Vibration tests should not be conducted for the sake of testing alone, but rather to obtain reliable systems. Thus, any vibration test which

Table 1

Summary of Cube Vibration with Excitation in One Plane

Excitation Plane	Acceleration Components					
	X	Y	Z	X'	Y'	Z'
X	10	0	0	6	2	8
Y	0	10	0	6	6	6
Z	0	0	10	6	8	2
X'	10	10	10	17	0	0

produces the desired responses in the specimen and discloses vehicle hardware defects, accomplishes this objective. Establishment of a single oblique-plane test to obtain the proper responses requires a complete knowledge of the specimen's internal construction to prevent over-testing in the oblique plane. The techniques of packaging design permit many packages to be tested by this technique. This method is probably limited to applications where there is a constant relationship, throughout the frequency range, between the desired acceleration vectors and a single monitor accelerometer.

Figure 3 shows an electronic package being vibrated in an oblique plane to detect workmanship and component defects with a minimum of flight hardware fatigue.

The single oblique-plane technique of conducting a multi-plane vibration test offers the advantages of reduced test facility costs, reduced test flow time, and reduced operating time of the specimen.

SIMULTANEOUS PRINCIPAL-PLANE TESTING

A technique using three vibrators to excite all three planes simultaneously is shown in Fig. 4. A separate vibrator is employed to excite each principal plane thus permitting the acceleration levels in each of three planes to be individually controlled as required. Oscillograph records taken with the three-vibrator setup verify that a single oblique-plane test is identical to a simultaneous three-plane test, provided the phasing is the same throughout the frequency range of all three exciters. Additional oscillograms show the ability to verify the input acceleration levels in each of the three principal planes. The records have been abbreviated and are shown in Fig. 5. As the

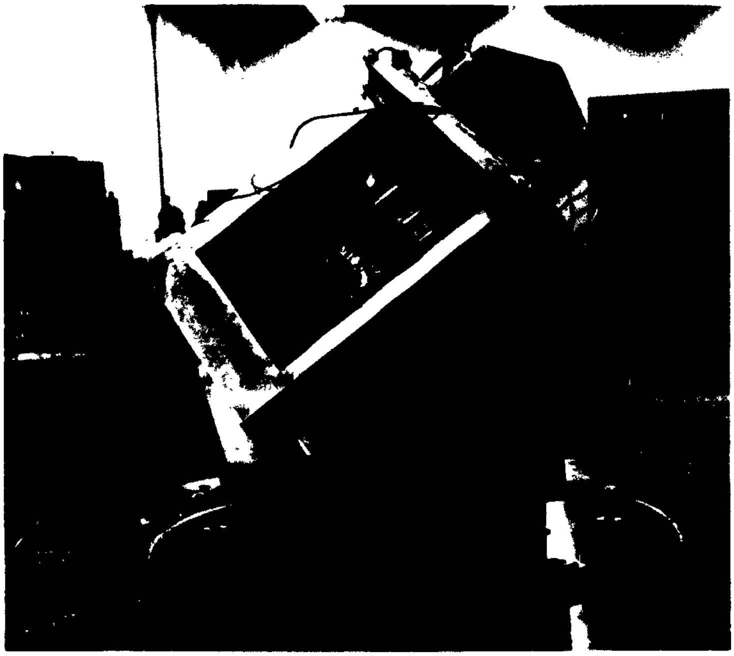

Fig. 3 - A single-plane vibration test through the diagonal of a cube

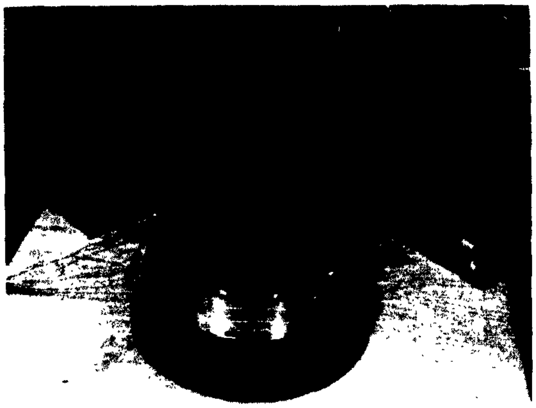

Fig. 4 - Test setup for excitation of a "cube" with three vibrators

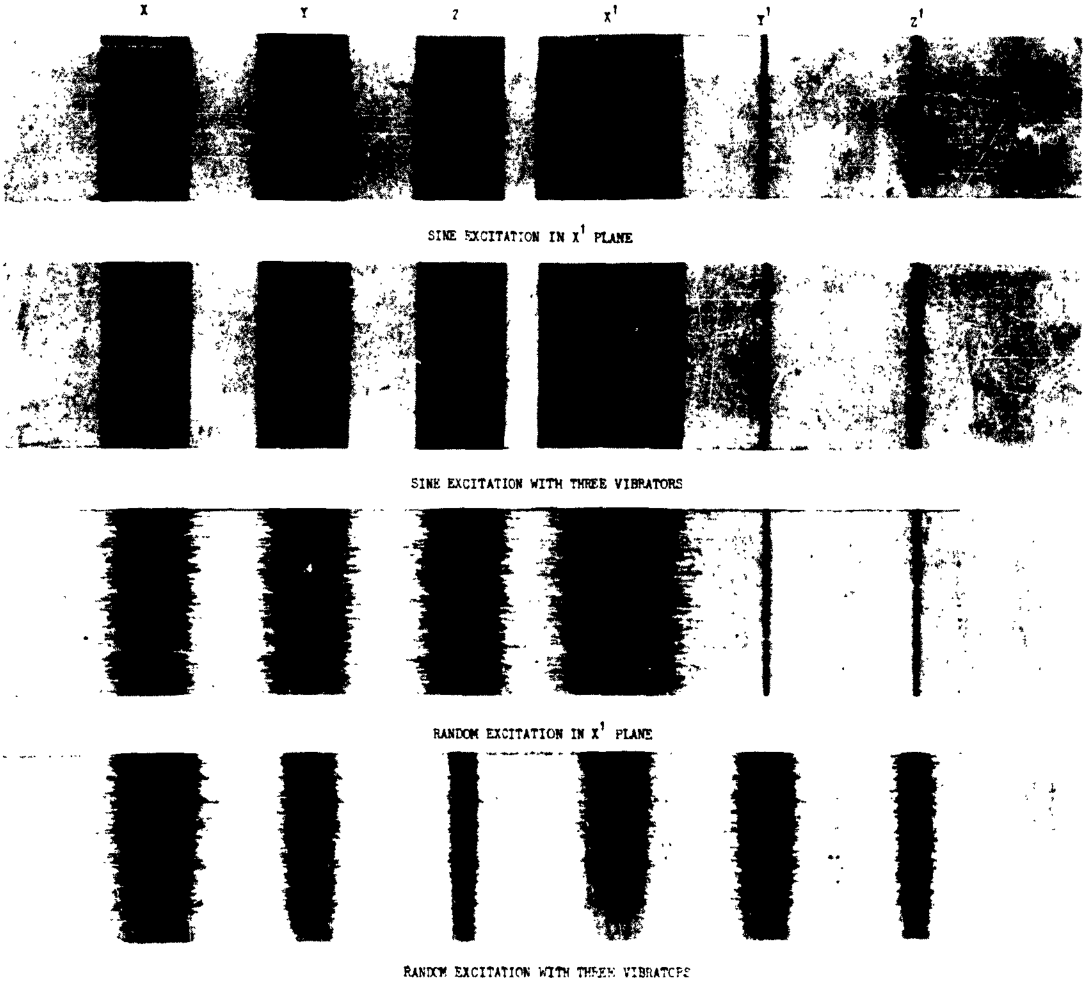

Fig. 5 - Response in six planes to sine and random vibration in one plane vs. three planes

specimen size increases from packages to whole systems, it becomes more and more difficult, if not impossible, to control the extraneous side motion of the specimen as required for the customary three-plane vibration test. Fig. 6 shows a test arrangement presently being investigated, which will reduce side motion with large specimen-to-armature weight ratio.

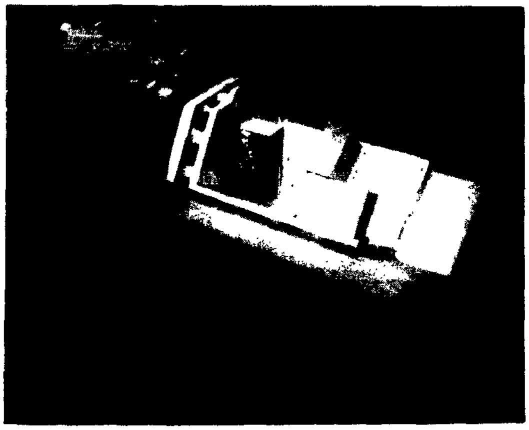

Fig. 6 - A system for vibrating complex specimens in each of three planes with a minimum of lateral motion

However, during simultaneous three-plane vibration tests, there is no undesirable side motion since total vibration is specified. In some instances there is sufficient vibration in an off-axis so that only limited additional force need be introduced. For example, the sine wave response, obtained from the flight monitor locations in a 400-pound system being vibrated in the vertical plane only, is plotted in Fig. 7. The off-axis vibration was equal to or greater than the specification levels throughout the major portion of the frequency range.

A test arrangement to make use of this off-axis vibration is shown in Fig. 8. The fixturing consisted of the vehicle mounting structure bolted to a stiff transition casting with a flat smooth bottom. This casting served as a sliding plate on an oil table which was bolted to a 25,000 force-pound vibrator. A 5000 force-pound vibrator was connected to the horizontal plane of the transition casting with two 30-inch drive rods. These drive rods had machined flexures to allow the specimen to move freely in the vertical plane.

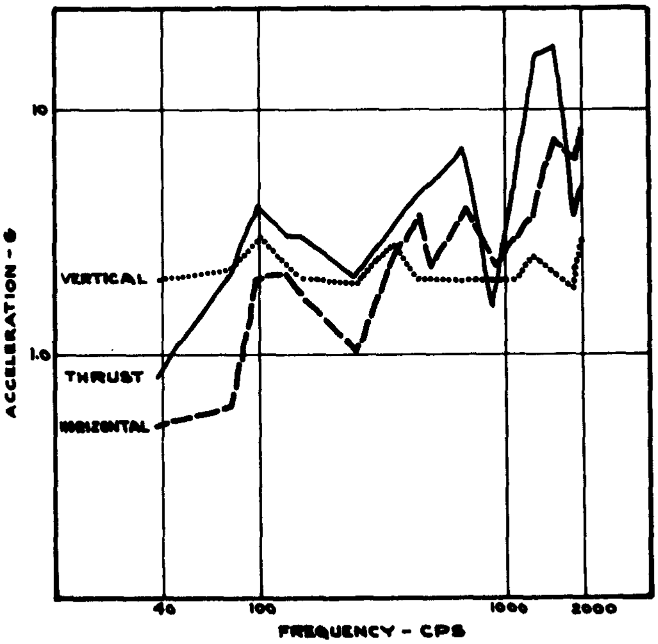

Fig. 7 - Frequency response of flight location accelerometers to excitation in the vertical plane

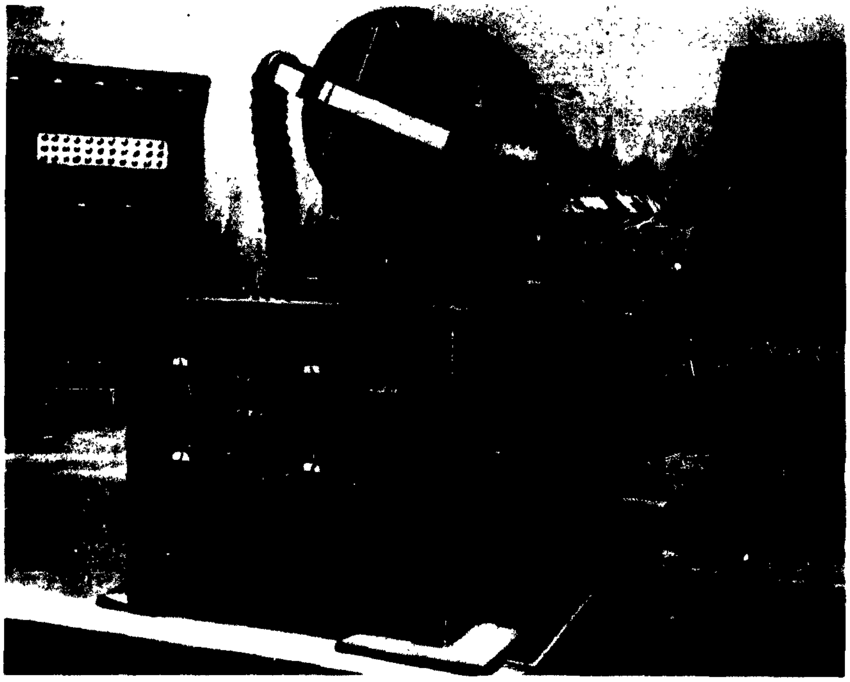

Fig. 8 - Test setup for excitation of a complex structure with two vibrators

For the test shown in Fig. 8, 24 accelerometers were monitored while the 3 flight monitor locations were used to determine the proper vibration inputs. The system was equalized in the vertical plane, with the second vibrator attached, by making a preliminary ±2-g sine wave sweep from 20 to 2000 cps, controlled by the vertical accelerometer only. From this information, the random wave input to the 25,000-force-pound vibrator was shaped to provide the specified vibration characteristics in the vertical plane. The specimen was vibrated for a few seconds while a sample of the vibration environment was recorded on magnetic tape.*

The analysis of the vertical, horizontal, and thrust responses showed that excitation in the three planes could be brought within the required specification limits by the addition of some low-frequency (20-500 cps) random vibration in the horizontal plane. The horizontal plane vibrator was then equalized in the same manner with a sine wave sweep. This time, both vertical and horizontal planes were vibrated simultaneously and the specimen responses recorded on magnetic tape. Figure 9 shows the analysis of the 3 flight monitor locations with excitation in the vertical plane only and with excitation in both vertical and horizontal after equalization was completed. A traveling sine wave which cycled between 50-100 and 300-2000 cps with both vibration systems, completed the vibration requirements for this specimen. The specimen responses obtained during this test correlated well with both flight and ground missile vibration data.

MULTIPLE SPECIMEN-PLANE TESTING

Figure 1, a test setup previously described for vibration transmissibility determination, is also illustrative of the simultaneous excitation of more than one specimen plane. In this case, the two shakers were not perpendicular to each other, and further, had no rectangular relationship to the principal axes of the missile. However, because of the use of the complete vehicle, very little equalization or spectrum

*For equalization and analysis techniques, see Ref. 1.

[1]Dubois, W. F., "Practical Random Vibration Measurement Technique," Shock, Vibration and Associated Environments, Bulletin No. 31, p.

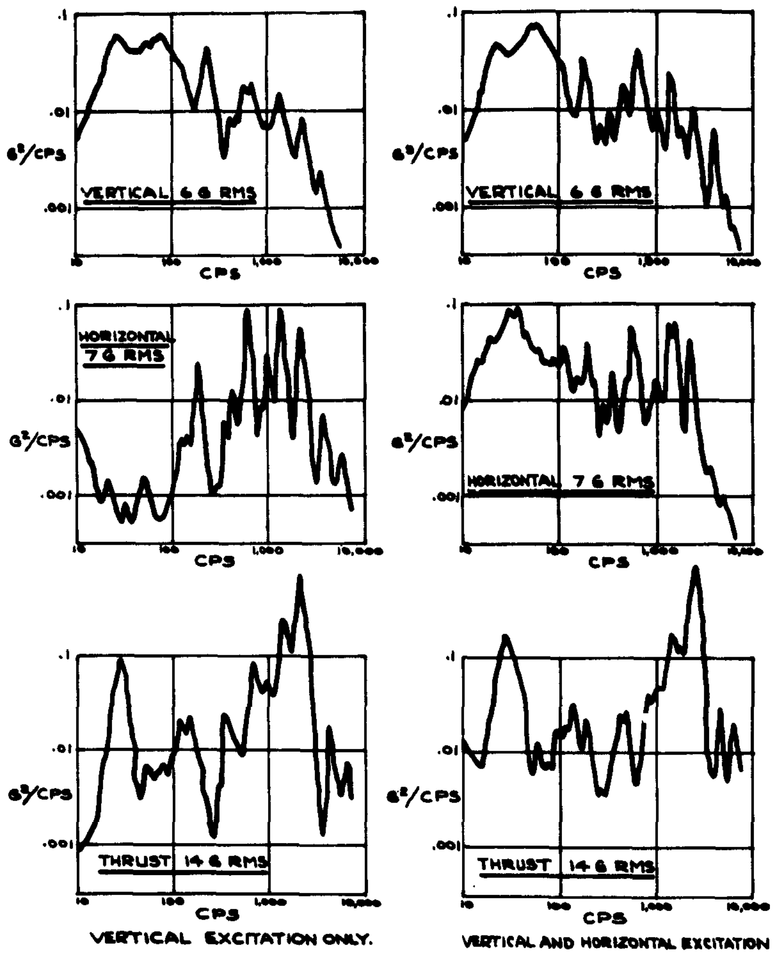

Fig. 9 - Response in three planes to random vibration input with one and two vibrators

shaping was necessary, since the overall missile structure had the effect of shaping the shaker inputs to the desired levels and spectra.

The test was used to isolate flight anomalies, which may have been caused by vibration, and to check the vehicle instrumentation.

CONCLUSIONS

The three methods of combining the three planes of vibration can be defined separately as follows:

1. Excitation in one of the many specimen planes to provide the desired vibration vector components.

2. Direct simultaneous excitation of the specimen's three principal planes.

3. Simultaneous excitation of more than one of the many specimen planes.

Before these techniques can be used for routine laboratory testing, further effort must be expended to determine the actual flight vibration characteristics and which vibration test can best provide the desired test results. Vehicle reliability improvements do not necessarily require an exact duplication of the service environments, but only an exact duplication of the service anomalies. The single vibration tests described can produce data to increase flight vehicle reliability at a reduced overall cost.

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DISCUSSION

Mr. Galef (NESCO): I would like to restrict my remarks to the case of the diagonal shaking. Actually if we look at the ordinary vibration test, we remember that we really are shaking in three planes all at once. They don't happen to be planes which are normal to the nominal axis, but why should they be? These vibration requirements are quite arbitrary, really. There is no particular point in saying that any particular axis is a true principal axis as far as the environment is concerned. If we want to look at what appears to be the geometry of the thing and say that these are the principal axes, that's OK. If we want to look at the records of actual flight and say that the total of some is along some other axis, that's OK too, but we really mustn't define a physical principal axis and start to shake along three planes normal to that until we actually have this data. I would also like to mention that this case of diagonal shaking seems to be completely identical to a paper that was presented at the 26th Symposium in El Paso by a man from GE, (Dr. Panariti).

Mr. Baber: He is referenced in the paper.

Mr. Stern (GE): I think this paper might have been labelled that a vector can be broken into three components. I think there is one thing you are overlooking in a lot of the testing. Even though you may drive a package along a principal axis and even though the motion may be fairly well defined, there is nothing to say that the component, if we were thinking of a printed board, is truely aligned in the same direction as the direction that you're driving. So even though you may use the three principal axes from a geometric standpoint, you actually excite practically all the components in all three directions even though you don't intend to. This is just a practical observation of what actually happens. In an actual test you excite components all the time no matter which way you are driving them. It's just that it is worse when you are exciting them in their weakest direction. An I-beam buckles in its weakest axis, not in its strongest. You can't predetermine its loading the way you would like it to be.

Mr. Baber: I probably didn't put my point across here, I guess. I have no qualms at all about the excitation in the three principal planes. I'm merely saying that that is what is usually done and the vibration tests are specified in these three principal planes. I think 90 percent of them are irrespective of any look at the internal structure and I am merely asking that the internal structure of the specimen be looked at — it has nothing to do with the principal planes — and that we come up with the actual motion that is desired on some specimen inside. The point I brought out was that, when we are running three separate tests, we are not taking into account the effects of the side motion. We sometimes monitor them, but we don't use them to reduce our test time and we also do not take into account the interactions that occur between the three simultaneous planes when they are being vibrated or when this item is mounted in the vehicle. You do not have an identical motion in all three planes, which is what we are doing.

Mr. Stern: Well, I get the impression you are suggesting a certain amount of investigation during the evaluation test of the particular black box to determine this. Now what I have in mind is that about that time when you are doing this, the product engineer is standing there and he has agreed to meet a contractual agreement for so many minutes in three planes, and he objects quite strongly. He will object to any of these additional investigations to determine what might be the best direction, or to anything you would like to do to his product, which of course has to be delivered at a time for a price.

Mr. Baber: Unless you can cut down his total test time.

Mr. Stern: This total test time is not up to him or you. This was signed in a contract 6 months ago when you agreed to deliver the product.

Mr. Baber: Well as Mr. Yeager pointed out, we should be getting into the original specs and not waiting until the job gets to the laboratory before we decide on the test.

Mr. Stern: One other point. How do you account for the fact that you induce three rotational degrees of freedom when you do what amounts to off-center-of-gravity testing? Do you concern yourself with this or are you not worried about it? There are 6 degrees of freedom, and if you don't drive through the cg there is going to be rotation.

Mr. Baber: All I'm asking is that the people are who writing the specs be aware of the fact that the laboratories can produce such motions on their item and that we would like to have them specify what these motions should be. These are some of the ways that we can actually produce this motion.

Mr. Nankey (GE): I would like to point out that the purpose of specifying testing in three mutually perpendicular directions is not to test those three directions, but to provide a certain minimum level of testing in all possible directions. This particular technique that you've described just fails to accomplish that purpose.

Mr. Baber: Except for one point, we are getting into the space environments. We are combining almost all environments with vibration and we may have an item that has, let's say, an operational life of perhaps 5 minutes. When we are trying to find out what is going to fail in this piece of hardware, we may have to produce 2 or 3000 degrees of temperature while we're vibrating; we may put this thing in the altitude chamber at some 200, 300, or 400,000 feet; we may produce the acoustic environment; and now, also, we have to run three separate vibration tests. It means that we either have to have three specimens, because they have a limited operational life or we have to decide at that time that we will pick one or the other plane or go to a technique similar to this.

Mr. Nankey: The point I was trying to make is that this is not adequate for exciting all the possible resonances in a given structure. Anything that is perpendicular to the direction that you have chosen to drive in will not respond.

Mr. Baber: But we can vibrate it with three vibrators if necessary.

Mr. Nankey's reply was inaudible.

Mr. Baber: Well, there is an infinite number of planes.

Mr. Forkois (NRL): We have been using an inclined bulkhead for shock testing for the last 15 years so if you want a reference this has been done for a good long while, well before the 26th Symposium. However the purpose of our using the 30-degree inclined bulkhead was mainly to solve a practical problem. Where you have items weighing 4000 to 5000 pounds and because the machine applied a shock motion in the vertical upward and downward directions, we had a problem of reorienting the equipment to apply shock motions in three directions. So to avoid doing this we mounted the equipment on a diagonal that was declined 30 degrees, much as you have done there, and in this way we were able to get shock components along three directions simultaneously. We were able, in this way, to produce damages which did not occur if items were tested in a single orientation only.

Mr. Schwabe (Lockheed): It has been my personal observation that the old-fashioned unidirectional tests tend to excite a frequency sensitive item at lower frequencies in directions orthogonal to the direction of testing. At higher frequencies an item is excited in many directions. I wonder whether you had noticed the same phenomena with your multidirectional testing?

Mr. Baber: You saw the one item that we had on there. It was a 400-pound specimen and it was being qualification tested. Now the point was that vibrating this thing three times, once in each of its planes, was infinitely more of a test than exciting the three principal planes simultaneously, or in any other method whereby we actually used the flight monitor locations and reproduced the same motion that occurred in flight on these three accelerometers. Again, all I want to point out, is that the tests are not run to obtain responses, they are not done just for the sake of vibration. The tests are supposed to be designed to detect flight defects and, in many cases — especially when we are talking about acceptance tests, we know where the weak spots are and we can orient our item to eliminate the need for a lot of the vibration testing that is presently being done.

* * *

HIGH INTENSITY SONIC TESTING—A TOOL FOR THE STRUCTURAL ANALYST

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Experimental evaluation of structural elements is the paramount factor in predicting the integrity of flight vehicle performance with respect to acoustically induced fatigue. Accurate prediction requires evaluation of the structural response, damping characteristics, system nonlinearities and a random fatigue test. A test technique is described which permits this evaluation to be performed in one versatile testing unit.

INTRODUCTION

The prediction of acoustically induced fatigue is strongly dependent on experimental evaluation and proof testing of structural designs. The analyst must interpret the fatigue history of representative examples of the structural configuration in terms of the fatigue life of the flight vehicle. Due to the complexity of the structural elements involved, the dynamic analyses required (prediction of resonant characteristics, localized stress levels, damping, non-linearities, and so on) are difficult and often impractical. For this reason, the structural engineer relies heavily on sonic fatigue testing of representative structural elements for basic design information.

The system required to perform this evaluation and proof test must be capable of generating sinusoidal and random noise over a broad frequency range. The use of sirens as an energy source permits either sinusoidal or random response, but can only in a limited way provide both. However, this flexibility is available in the electro-pneumatic transducer which generates acoustic power from pneumatic power by electrodynamic control. This device and its use in progressive wave testing for sonic environmental simulation will be described.

PROGRESSIVE WAVE STRUCTURAL TESTING

Progressive wave testing offers the advantage of compact test equipment size with the capability of generating wide band acoustic energy over the entire audio spectrum. Basically, the progressive wave system (Fig. 1) consists of:

- Acoustic energy source; electro-pneumatic transducers.

- Acoustic impedance matching of source to test section; a suitable exponential connector is used to match the small cross sectional area of the transducer to the larger test section area.

- Test section; a narrow channel is used with one side opened to accommodate the structural element to be tested. A square cross section is provided for electronic equipment evaluation.

- Acoustic termination; absorptive wedges are placed in a matching channel in order to avoid standing waves.

When structural testing is the prime requirement, a narrow channel is used to generate grazing incident sound. For example, the test section of the system shown in Fig. 1, is 6 by 30 inches. It will accommodate a panel with maximum dimensions of 30 by 40 inches. Acoustic energy is provided by four electro-pneumatic transducers which produce in excess of 8000 acoustic watts of power.

ELECTRO-PNEUMATIC TRANSDUCER

The principle of operation of an electro-pneumatic transducer has been known for at

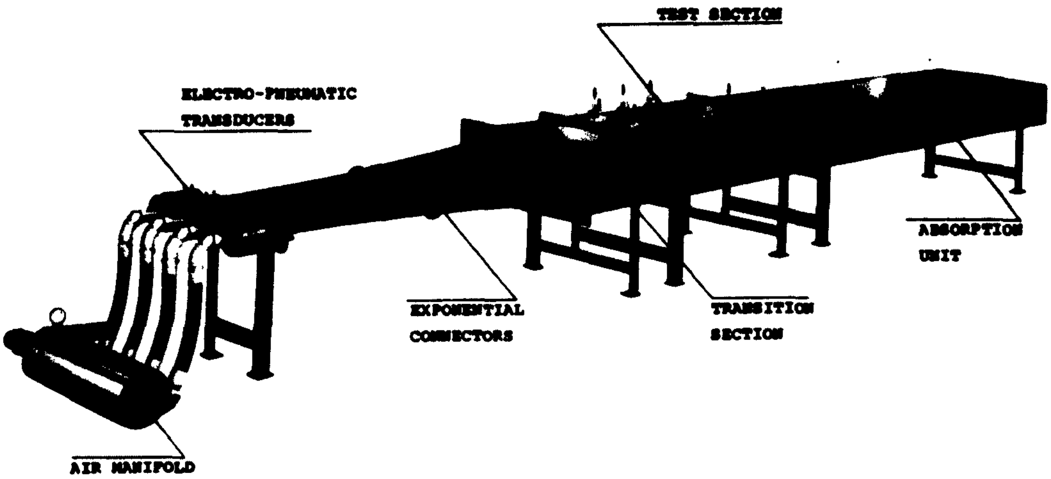

Fig. 1 - Progressive wave tube

least 25 years. A comprehensive review of its development is given by Burgess and Salmon.[1] The device now in use, illustrated in Fig. 2, is described in detail by Hilliard and Fiala.[2] Essentially, it consists of a stationary inner ring and a movable outer, modulator ring. Both the inner and outer rings contain two rows of matching slots. The outer ring is driven electrodynamically relative to the stationary inner ring and modulates air passing through the slots. At full modulation the device generates 2000 acoustic watts. This is accomplished with 300 scfm (standard cubic foot per minute) of air at 40 psi. The electrical power necessary to obtain 100-percent modulation is 100 watts. The response of the unit is flat to 800 cps and falls off at 6 db per octave above this frequency.

MODULAR DESIGN CONCEPT

With the electro-pneumatic transducer a modular concept in system design is possible. Each module consists of the transducer and an exponential connector which terminates in a 7 by 7-inch cross section. The exponential connector is 6 feet long and has a lower cut off frequency of 40 cps. This module is shown in Fig. 3. Various arrays can be constructed by use of these modules. An example of a 2 by 5 transducer array is shown in Fig. 4. With this array, 20,000 acoustic watts may be generated. Since the array is portable it can be used in either a square cross section progressive wave tube for electronic equipment evaluation or a progressive wave tube channel for fatigue testing. Also, a suitable connector may be provided for coupling the array to a reverberation chamber. The facility shown in Fig. 5 is being constructed at the Douglas Aircraft Company and exemplifies this possibility. In this facility, progressive wave testing to 170 db and reverberation chamber work to 160 db will be possible.

PROGRESSIVE WAVE CHANNEL FOR STRUCTURAL EVALUATION

A four-transducer progressive wave channel is in use at the Research Center. An overall view of the facility was shown in Fig. 1. The progressive wave propagates to a 6 by 30-inch cross section in which structural elements having maximum dimensions of 30 by 40 inches, may be tested. Figure 6 shows a panel in the test section of the tube; it also illustrates the quick disconnect clamping arrangement which provides easy access to the panel for inspection.

CONTROL AND ANALYZING EQUIPMENT

The schematic of the complete control and analyzing equipment in use with the progressive wave tube described is shown in Fig. 7.

[1] Burgess, J. C. and Salmon, V., "Development of a Modulated Air Stream Loudspeaker," Stanford Research Institute (Dec. 30, 1955).

[2] Hilliard, J. K. and Fiala, W. T., "Methods of Generating Increased Sonic Power For Environmental Testing," IES 1960 Annual Convention, Los Angeles (Apr., 1960).

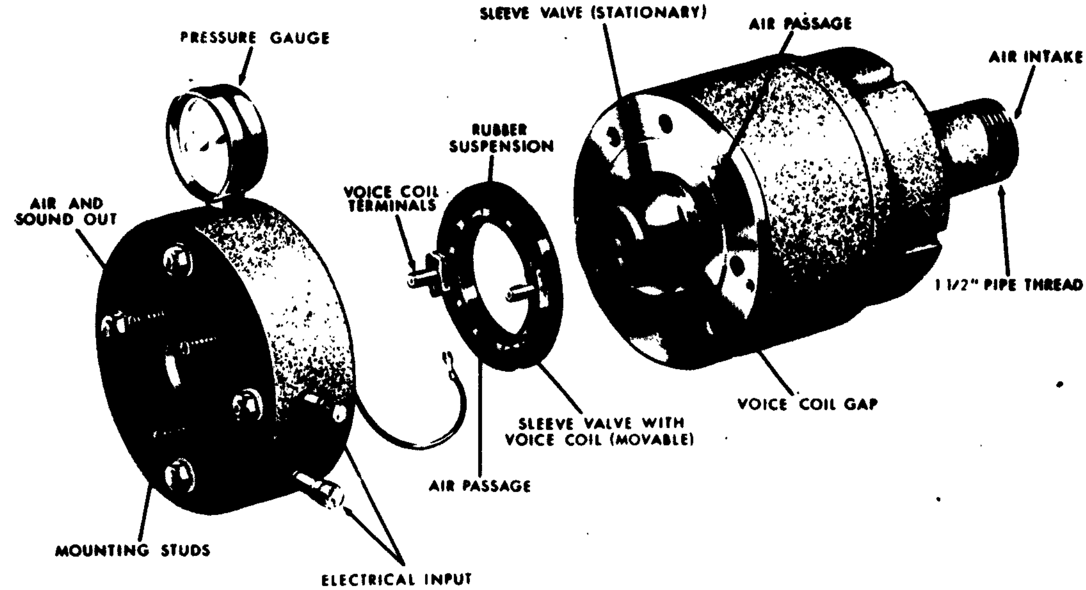

Fig. 2 - Electro-pneumatic transducer assembly

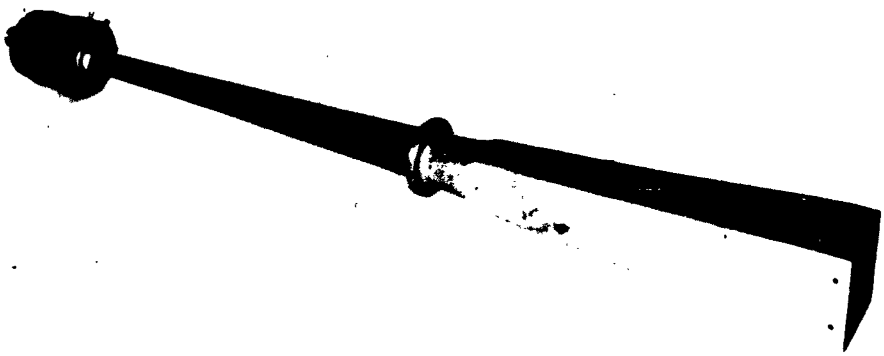

Fig. 3 - Transducer/connector module

Fig. 4 - Transducer/connector array

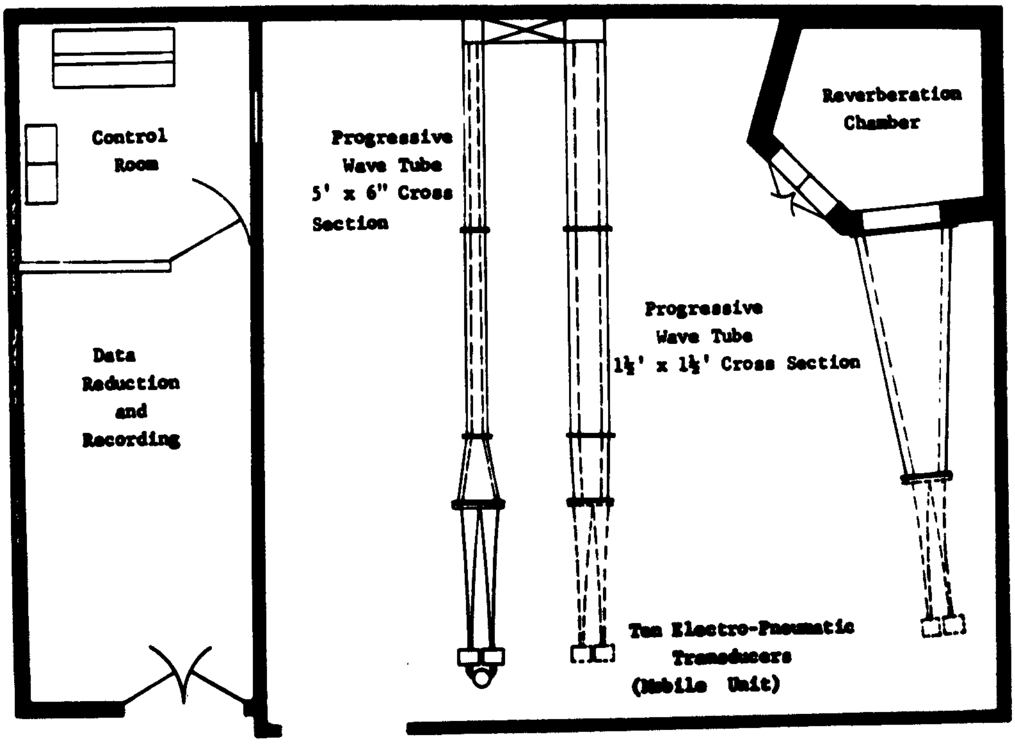

Fig. 5 - Sonic environmental testing laboratory

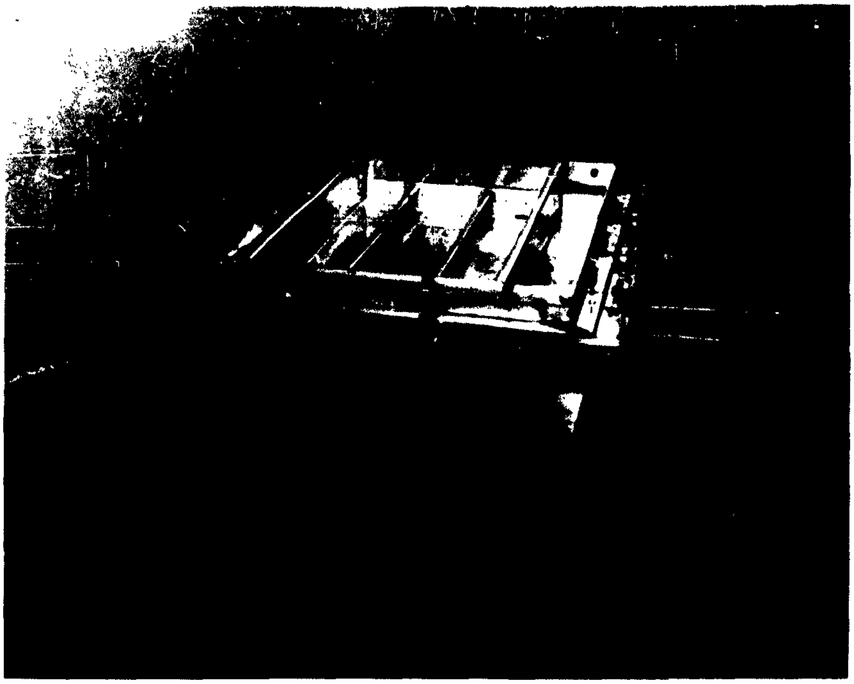

Fig. 6 - Typical structural panel in progressive wave channel

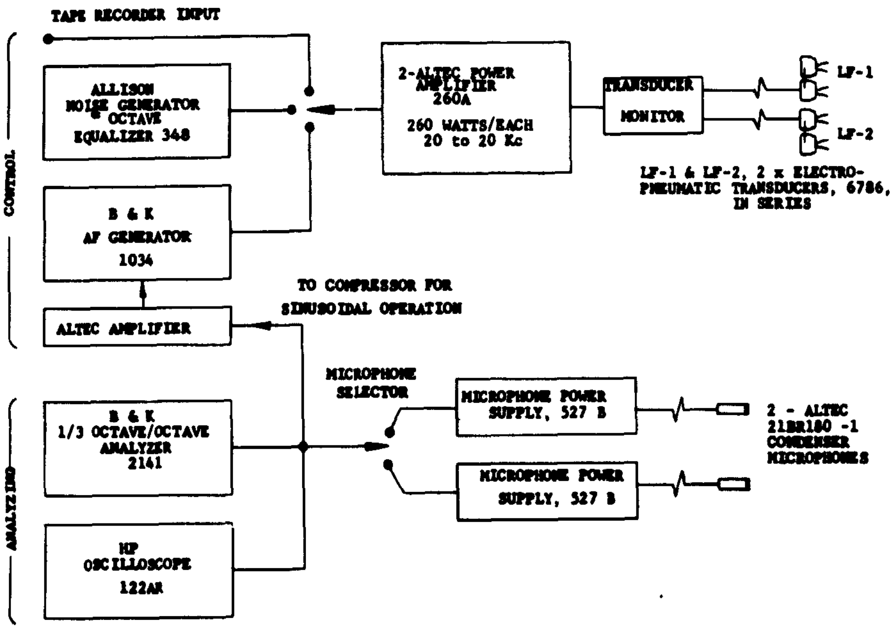

Fig. 7 - Control and analyzing equipment for sonic environmental system

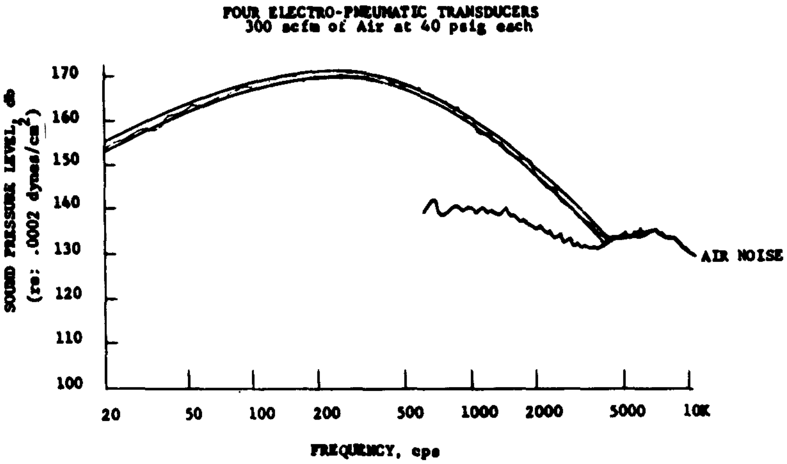

Fig. 8. - Frequency response of 6 by 30-inch progressive wave tube

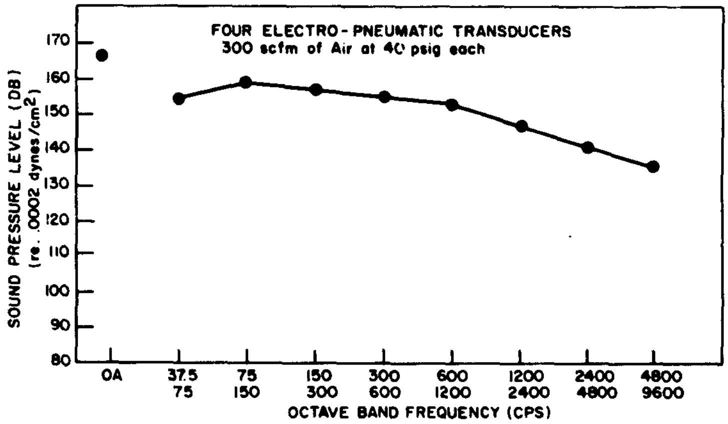

Fig. 9 - Random noise spectrum of 6 by 30-inch progressive wave tube

Three signal sources can be selected: a white noise source, a sinusoidal signal, or the output of a magnetic tape recorder. Shaping of the white noise signal is possible with the octave equalizer which allows individual attenuation of each band. Measurement and analysis of the pressures in the test section are accomplished with condenser microphones and a spectrum analyzer.

The condenser microphone output may be coupled to the compressor input of the audio frequency generator by means of a preamplifier. With this arrangement, the sound pressure at the selected microphone position can be kept constant throughout the sweep range of the generator within the system capability.

With the electro-pneumatic transducer operating at full modulation the maximum sound pressure response, shown in Fig. 8, was obtained. In general, approximately 5-percent harmonic distortion occurred at these maximum levels. With an exponential connector having a lower flare constant, adequate response could be obtained down to 5 cps. This has been

demonstrated in our laboratory in a 1 by 1-inch progressive wave tube. The maximum random noise environment generated in the 6 by 30-inch progressive wave tube is shown in Fig. 9. An overall sound pressure level of 166 db (re: 0.0002 dynes/cm^2) was achieved with an octave distribution as shown in Fig. 9. At these high sound levels, approximately 8-db side band octave control of random noise is possible. At lower levels, much greater side band control can be achieved. The distortion of the signal as it propagates down the tube creates harmonics and provides significant acoustical energy above the normal operating range of the transducer.

CONCLUSION

By use of progressive wave testing and the electro-pneumatic transducer, it is possible to provide an energy source for a wide range of testing situations. It meets adequately the requirements of structural fatigue testing, providing both sinusoidal and random noise in one unit. Considerable growth is possible as a result of the modular concept in source design.

DISCUSSION

Mr. Schwabe (Lockheed): I noticed that you had a heavy type of coating on those horns. Could you explain what that particular coating is?

Mr. Van Houten: That's a material called aquaplaz. It is made by Blanchford Corporation in the east. A fairly gummy substance when it goes on, it gets relatively hard, but has very good damping properties. The tube that you saw is of double walled aluminum construction and both sides of the inside plates also are coated with 0.25 inch of aquaplaz.

Mr. Volin (NOL White Oak): What is the noise level on the exterior of the first test unit where people would walk around? How would you describe it?

Mr. Van Houten: Very unpleasant. This type of a tube, particularly a tube for structural testing, would have to be placed in an abatement room away from the control area. This is mainly because your structural panel is the weakest link acoustically in the system and is going to provide at best, for a normal missile or aircraft type structure, 20-db attenutation. So if you are generating 165 db inside you've got 145 db outside, the chamber plus reverberation buildup outside the chamber. So you do have to put this small facility in an abatement room.

Mr. Volin: How does the noise level fall off as you get away from the chamber? I assume that you have some attenuation as you move away from the chamber. I'm speaking of cases where you have to run a full power test.

Mr. Van Houten: Essentially, the abatement room we constructed at our facility consists of two, 2 by 4 isolated walls, rather an inexpensive and temporary type structure. At full power operation, 166 db, we had about 90 db in our test area which was within 6 feet of the tube, but isolated by this wall.

Mr. Hillyer (Sandia): Can you give me some idea of the noise gradient as you go away from your 170-db level, are you measuring this right at the center of your test section at the start or what?

Mr. Van Houten: The response curve shown is with a dummy, extremely heavy, stiff panel in the test section. The panel greatly affects the response of the progressive wave as it propagates down the tube. I would say there is certainly going to be a considerable gradient. There is a very good Southampton report on such a facility that does describe the spatial correlation within such a test section. I think your question has to do with spatial correlation. They cover this aspect of the problem quite thoroughly.

Voice: Do you get any cross talk?

Mr. Van Houten: I believe the question is do we have standing waves perpendicular to the chamber? Certainly you do. As we go up in frequency, we are going to drive all the eigentones or modes of the chamber. Theoretically, the volume velocity is in a direction which doesn't permit a most advantageous drive of these cross modes, but they do exist.

Mr. Gauvain (Atomics International): Do I understand that this horn will give you 10 kc?

Mr. Van Houten: No, I described the response of the device as being flat to around 800 cycles and then falling off at about 6 db per octave, except that it goes off quite steeply at around 2 kc. It's an electrodynamic device and is mass controlled at the high end. The problem

is in getting the frequency high enough while still being able to move the small modulator ring with an amplitude sufficient to modulate the air stream.

Mr. Morrow (Hayes International): Can you get precise control of your spectral shape?

Mr. Van Houten: At low levels, let's say, in the order of 145 to 150 db, yes you can. You can put in a third octave and you get out the third octave. As you go on up in frequency the propagation distortion is generating significant harmonic energy and, at about 160 db, I would say your side band octave control is of the order of 8 db.

Mr. Morrow: Can you shift your peak to fairly low frequencies?

Mr. Van Houten: Certainly. As I say, it is an electrodynamic device. It will modulate at the frequency at which it is excited, particularly at low frequencies, since below resonance it's relatively flat.

Mr. Euler (Bendix): Do you calibrate your own microphones, and if so, what is your estimate of accuracy at 170 db?

Mr. Van Houten: The calibration of the microphones is performed by the Altec Lansing Corporation, as you might suspect. I would say the accuracy in this range is quite good - plus or minus 1 db. A condenser microphone is used; this is a relatively easy device to calibrate at these higher sound levels.

Dr. Fricke (Bell Aerosystems): You pointed out your acoustic power is about 2000 watts, is that correct?

Mr. Van Houten: Yes sir.

Dr. Fricke: How did you measure your acoustic power?

Mr. Van Houten: This is pretty easy and quite accurate in a progressive wave tube because you are measuring the sound pressure level across the tube. From this you can convert directly to intensity which gives you the wattage per square centimeter. You know the number of square centimeters, so you can calculate the power.

Dr. Fricke: This is right as long as you don't assume any loss between your output and the point where you have your microphone. How good is your progressive wave system? I guess you might have some reverberant characteristics, so do you have a complete absorption in your progressive wave system? How did you control this? It is a pretty important point because sometimes you like to have a reverberant system to simulate reverberant conditions and sometimes for acoustical fatigue tests you like to have progressive wave system.

Mr. Van Houten: You can certainly get an idea of this directly from a response curve, particularly at lower levels, where you don't have the distortion. As you noticed the response of the device is falling off starting at around 100 cycles. We noticed a very definite standing wave at around 40 cycles with an antinode occurring around 60 cycles, so that this is causing our fall off. The answer to your question is that we did not have a good progressive wave at these lower frequencies. The reason is the limitation on the absorption unit. You can have as good a standing wave as you want to make an absorption. The longer you make the absorption, the better progressive wave system you are going to have. There are some practical limitations here on the length of the tube.

Mr. F. Edgington (White Sands): Asked how input to the modulator was controlled to obtain the octave band distribution of Fig. 9?

Mr. Van Houten: Actually, what was done here was equalization through the octave equalizer to give the maximum output of the transducer. Putting in a constant level of all frequencies is inefficient since then the devices would be over modulating at low frequencies and under modulating at higher frequencies. So you tend to put the energy in at the higher frequencies.

* * *

THE USE OF A VACUUM TECHNIQUE FOR ATTACHING A TEST FIXTURE TO A VIBRATION EXCITER

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In a testing program where a test is to be repeated many times it is expedient in both cost and schedule to design a test setup that can be accomplished in the minimum time. This paper presents a method for attaching a vibration fixture to an exciter by means of a vacuum suction rather than the conventional bolt-tapped hole method.

INTRODUCTION

The production test requirements on the MINUTEMAN Stage I, II, and III hydraulic servocylinders, included a vibration test on each servocylinder of 5-g rms from 5 to 1000 cps. The number of items to be tested during the first phase of the program was 600 for each stage in each of 3 axes — a total of 5400 axes.

The test procedure required that the input acceleration at the three attach points of the servocylinder be at the same intensity within ±1-g rms, therefore it was necessary to design a very rigid fixture with low transmissibility over the vibration frequency range. Three major disadvantages were encountered in a design using the conventional method of bolting the fixture to the vibration exciter. First, since it would have been necessary to remove the servocylinder and all electrical and hydraulic connections from the fixture to gain access to the bolts, excessive time would have been required for this operation. Second, the bolt holes in the fixture would have worn severely before the end of the production program. Third, it would have been difficult to have tightened the bolts each time so as to insure repeatable acceleration inputs to the servocylinder. These problems were eliminated or greatly reduced by use of the vacuum suction technique. In this technique, a small pump is used to pull a vacuum between a permanent plate bolted to the exciter armature and the jig fixture. The suction created by the vacuum was sufficient to hold the fixture in place. Changing axes was accomplished simply by turning off the vacuum pump and physically lifting the fixture from the permanent plate and rotating it to the next desired axis.

FIXTURE DESIGN

Design Objectives

The primary design objectives were:

- To minimize fixture resonances and transmissibility so that the specified input acceleration at the three attach points could be realized with acceptable test repeatability;
- To minimize the time required to mount the fixture on the exciter and to change its orientation from one axis to another;
- To keep the total weight within the capability of the exciter without using external flexures; and,
- To minimize the common problems with bolts, such as stripping or wearing out hole threads and excessive wear of fixture holes.

Vacuum Technique Theory

The force, F, holding a cap on a vacuum chamber is given by

$$F = AP,$$

where A is the area of the cap, and P is the pressure differential on opposite sides of the cap.

It was realized that if a vacuum chamber could be fabricated and bolted to a vibration exciter, such that the chamber cap would be a part of the vibration fixture, an expedient way of mounting the fixture to the exciter would be obtained. In the final result, a circular cap with a 6-inch diameter was used. The vacuum pump provided a pressure differential of approximately 14 pounds, therefore the force holding the fixture to the exciter was approximately 396 pounds. The greatest mass involved during the test, including fixture, vacuum chamber, and test article, was 26 pounds. The peak force acting to separate the fixture from the exciter is given by

$$f = 1.414\,GM,$$

where G is the rms acceleration and M is the mass of the cap, fixture, and test article. Since $G = 5$ and $M = 26$ the peak force was 184 pounds. Thus the force margin was 115 percent.

Basic Fixture

Since the Temco Test Laboratories have had outstanding success with K1A magnesium castings in the fabrication of production vibration test fixtures, the basic fixture was cast from this material. The fixture for the Stage II servocylinder is shown in Fig. 1. It was necessary to lighten the Stage I and Stage II fixtures by drilling weight reducing holes in the 2-inch thick walls. Each fixture accommodated the servocylinder for vibration in all three axes. Instead of drilling and tapping directly into the magnesium for the test article mounting bolts, steel inserts were installed in the fixture. These bolts not only added strength, but prevented thread deterioration as the bolts were repeatedly inserted and removed during the production program. The fixture overall dimensions were approximately 11 by 9 by 6 inches.

Receptacle

An aluminum receptacle, illustrated in Fig. 2, was bolted to the vibration exciter armature. Receptacle depth was 0.50 inch. The inside diameter was 6.000 +0.001, -0.000 inches. The diameter of the bottom of the receptacle was that of the armature, 4 inches. The outside diameter of the receptacle was 6.75 inches. It was necessary to make the diameter of the receptacle greater than that of the exciter armature to provide sufficient vacuum area to generate the force required to hold the fixture in place and assure stability. The height and depth of the O-ring groove was 0.170 and 0.123 inch, respectively, as suggested by the manufacturer of the 6 by 6.25 by 0.125-inch Buna N rubber O-ring used. A hole was drilled in the bottom and side of the receptacle to accommodate the line to a small vacuum pump. A two way valve in the vacuum line provided an expedient means for venting the atmospheric pressure to the vacuum chamber during the process of changing axes.

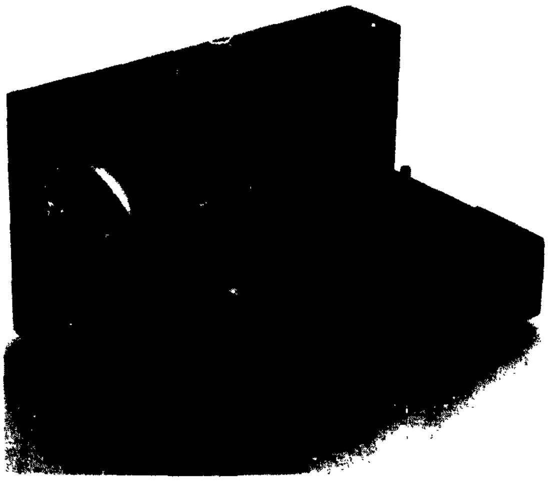

Fig. 1 - Basic fixture

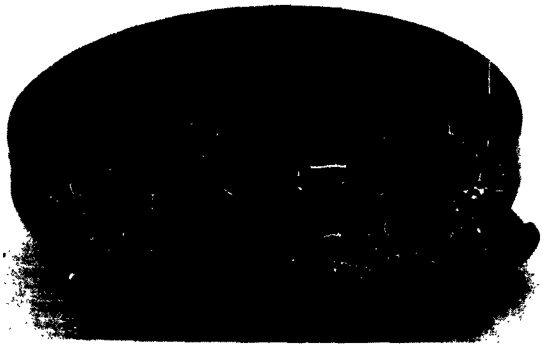

Fig. 2 - Seal cap

Aluminum was chosen for the receptacle and seal caps since it was better suited than magnesium for rough handling by the test personnel. The critical area of the basic fixture was quite well protected by the seal cap.

Seal Cap

An aluminum seal cap 5.994 + 0.000, -0.001 inches in diameter was bolted to each face of the basic fixture, as shown in Fig. 1. The thickness of the cap was 0.40 inch, which was 0.10 inch less than the depth of the receptacle. It was desirable to have as small a volume as possible between the cap and the receptacle so that minimum time would be required to create the vacuum, yet be assured that in the final assembly the cap would not touch the receptacle. The only sealant required between the fixture and the cap was a heavy silicon grease. Possibly any other grease would have worked as well. Care was taken in locating the caps so that the center of gravity of the fixture and test article would align with the center of the vibration exciter. It might be pointed out that in one case the cap edge extended approximately 1/4 inch beyond the edge of the fixture. This evidently caused no resonant problems during the test. The only force acting on the cap was the force generated by accelerating the cap.

Fig. 3 - Complete assembly mounted on vibration exciter

Final Assembly

When the cap had been inserted in the receptacle, the face of the fixture rested on the upper rim of the receptacle. After the caps had been attached to the fixture, a course lapping compound was placed between the fixture and the receptacle rim. The surfaces were ground by hand, approximately 5 minutes each, until the mating surfaces were flat. A course lapping compound was used since a high finish was undesirable because any grease or oil between the two high finished surfaces would have decreased the ease of separating the surfaces. Figure 3 shows the complete Stage I servocylinder assembly mounted on the vibration exciter. Figure 4 shows the complete test setup.

Fixture Evaluation

Preceding the test, each fixture was loaded with a dummy mass and the vibration level at which the fixture and receptacle became separated was determined. The maximum level without chatter ranged from 10- to 14-g peak for the various fixtures. For one or two of the eight fixtures the g level was less than 10, however, additional fixture-receptacle mating surface grinding increases the point of separation to at least a 10-g peak.

The evaluation of the fixture indicated achievement of the design objectives. Maximum total weight was 26 pounds, well within the capability of the exciter. The time required to mount the fixture, after the servocylinder had been installed, or to change axes was approximately 10 seconds. There was no repetitious bolt operation involved in the setup, except that required to mount the servocylinder to the fixture.

COMPARISON OF VACUUM VERSUS BOLT TECHNIQUES

During the quality verification tests (QVT), which were performed simultaneously with the production tests, duplicate basic test fixtures were used. Since these QVT tests required a much higher vibration test level, the vacuum method could not be used; instead, the fixtures were bolted to the vibration exciter. Therefore, since the setups for qualification and production tests were identical except for the mechanical coupling, a direct comparison of time and fabrication costs of the two methods was obtained. The following expenditures do not include costs common to each method such as procurement, design, and machining of the basic fixture. The cost of two receptacles, 24 caps, sealant, and assembly, including all material and labor, was approximately $1000.00. The vacuum pump cost an additional $100.00. Since this method was a "first effort" for Temco, engineering and design time was 24 hours, which is greater than it will be in the future. The total cost for eight complete vacuum type assemblies was approximately

Fig. 4 - Complete test setup

$1500.00, or less than $200.00 each; whereas, cost for preparing four bolt type fixtures was approximately $460.00, or $115.00 each.

Time required to change axes, or to change from one test article to another, during the qualification test, using bolts, was about 1 hour; whereas the vacuum method required about 10 seconds. Thus, the saving was approximately 5400 manhours or, with two 10-hour shifts per day, 270 days on the production schedule. Of course, this schedule time could have been made up by procuring additional vibration exciter systems and using more technicians on the production test.

Proper consideration should be given to the fact that a large excitation caused by an error by the operator or an exciter malfunction could cause the fixture to become separated from the receptacle. Some means of preventing the fixture from falling to the floor, if separation did occur, should be provided. For the application presented here, the hydraulic lines provided sufficient support to protect the servocylinder.

Errors resulting in an excitation level above approximately 10-g peak are quickly detcted because the operator can hear the fixture-seal cap clatter. This is another advantage of the vacuum method, particularly when less experienced technicians are performing the tests.

APPLICATION HISTORY

The fixtures and vacuum mount were completely satisfactory after 8 months of continuous use. The only replacement parts required were a few O-rings. The receptacles and caps experienced a few scratches and dents, but continued to operate properly.

There were only three occasions when the fixture became separated from the exciter due to transients in the exciter system.

FUTURE APPLICATIONS

The vibration system used for the MINUTEMAN servocylinder production test consisted of a Ling Model 219 exciter and a Ling RP 3/4 amplifier. Two systems were used during the program. The vacuum method could be used on larger exciters for certain tests. Table 1 shows the force available for holding the fixture for a few exciters if the receptacle diameter is the same as the shaker head. If the diameter of the receptacle is extended by 4 inches in a manner similar to that of the MINUTEMAN setup, the forces would be as in Table 2.

Some modifications for the larger diameter fixtures would be necessary to decrease the problems of resonance. One suggestion would be to fabricate the receptacle with one or more rings in the bottom to support the center portion of the plate. It would be necessary to grind all mating surfaces at the same time to insure an even support between the two parts.

TABLE 1

Shaker		
Model No.	Head Diameter (inches)	Holding Force (lb)
A175	8.875	870
A246	16.250	2900
A249	28.375	8850

TABLE 2

Shaker		
Model No.	Head Diameter (inches)	Holding Force (lb)
A175	12.875	1820
A246	20.250	4500
A249	32.375	11500

* * *

COMBINED ENVIRONMENTAL TESTING ON THE HOLLOMAN TRACK

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Deputy for Guidance Test
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The combined vibration and sustained acceleration environment of the Holloman Track have pointed out system failures which had not been detected by sequential laboratory tests. The capabilities of the high-speed track and the favorable results of the track testing are discussed and a number of examples are given.

INTRODUCTION

The problems inherent in the application of environmental data to systems specifications and design criteria are well recognized and have been the subject of much thought and profound study. Equally important, but perhaps less well defined, are the problems of determining the success of such an application before the system is called upon to perform. While it is convenient to be able to say that statistically we have an 80-percent confidence that our system is 99.9-percent reliable, the facts are that in the aerospace industry the quantity of fabricated systems is statistically very small. Our confidence and reliability figures are based upon a statistical manuever which attempts mathematically to relate the laboratory determination of component reliability to reliability of the entire system. That this procedure has lead to overdesign of components and has failed to accurately predict systems reliability is a matter of record. What is needed then is a method whereby the system can be tested, recovered, modified, and tested again in an environment representative of, but exceeding, the planned environmental modes of operation. Only in this manner can true system reliability be determined with any degree of confidence.

Fortunately, such a method is available. The high speed, 7-mile, test track of the Air Force Missile Development Center at Holloman AFB has for some time been engaged in the development of techniques, facilities, and equipment for the purpose of testing systems and components in an environment representative of that met in aerospace operations. The purpose of this paper is to acquaint the reader with this facility and to illustrate the advantages of its use in combination with standard laboratory test techniques.

THE TRACK AND ITS EQUIPMENT

Figure 1 is an aerial view of the south end of the track looking north and shows a portion of its 7-mile length. Two continuously welded steel rails 7 feet apart are aligned and ground to a tolerance of ±0.01 inch. The vehicles carrying the test item are capable of velocities from a few hundred feet per second to Mach 4 and of accelerations to 100 g. Generally these vehicles are of four types.

The general purpose vehicle (Fig. 2), is propelled by a liquid engine developing 120,000 pounds of thrust. It has a peak velocity of Mach 1.5 and can be programed for accelerations or decelerations or both of up to 8-g with a payload of over 1 ton. It is capable of providing a systems environment closely related to the first stage of missile or space vehicle flight. A more specialized vehicle (Fig. 3) is used for testing radomes, nose cones, and so on, in a rain and vibration environment up to speeds of Mach 3. The rain drops are controllable in size and the quantity of water can be programed for up to 8 inches per hour along 6000 feet of track. Another specialized vehicle (Fig. 4) is the monorail sled used for impact and vibration tests of fuzing systems. Used as an impact vehicle it is capable of speeds to Mach 4. The most specialized vehicle (Fig. 5) is the one designed and fabricated by Holloman personnel. This particular vehicle is used to test the Dyna-soar landing gear under operational conditions

Fig. 1 - Air view of south breech

on various types of surfaces. In addition to these, many other types of test vehicles are available, all capable of producing environments tailored to the needs of the system or component under test.

TRACK ENVIRONMENTAL DATA

The end product of such tests is, of course, data. Such data can then be used to determine not only the success or failure of the system under test, but also to determine what failed and most important where, why, and under what conditions failure occurred. This information may reasonably, take the form of visual data such as that shown in Fig. 6. Examination of the test item (in this case a missile nose cone undergoing rain test) leaves no doubt that failure occurred. Further examination of the accelerometers mounted on the bulkhead reveals complete failure due to rain impingement after failure of the nose cone. It is interesting to note that had these accelerometers, as part of a fuzing system, had a reliability of 100-percent the system would nontheless have failed since resistance to rain impingement is not normally one of their design specifications. Information in the form of charts and printed sheets is also available from telemetry data, see Figs. 7 and 8. This data are in turn related directly to Fig. 9 which shows the velocity of the test vehicles with respect to time. Although Figs. 7 and 8 are representative of measurements in one direction at one point only, the final test report presents similar data from many points and in all important directions so that the three directional shock, vibration, and time history of the test item is known throughout the test.

UTILITY OF TRACK TESTING

Significantly, the technique of using the new high speed track as a method for determining

Fig. 2 - Acid engine

Fig. 3 - Rain erosion sled

systems reliability began to develop during the track tests of inertial guidance systems, which were also relatively new and, heretofore, subject primarily to environmental laboratory testing of components. As is well known the primary purpose of guidance system track testing is to obtain an error model. However, an analysis of guidance test data revealed that

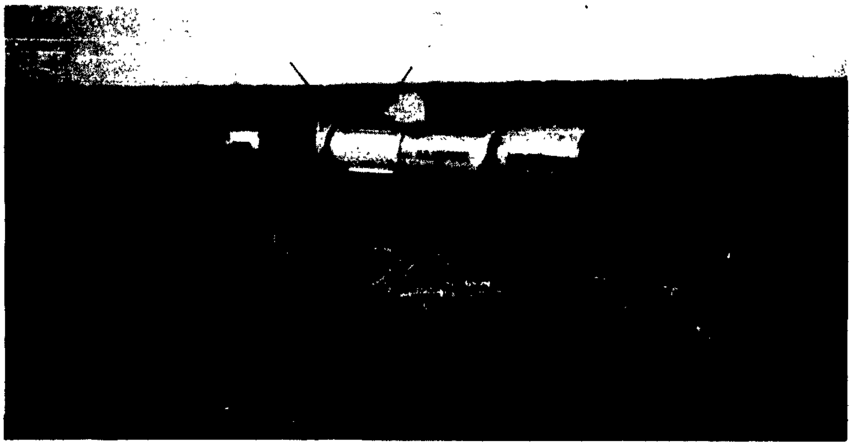

Fig. 4 - Impact sled

despite the most rigid design specifications, controlled fabrication, and advanced laboratory environmental tests of components, systems failures were frequently caused by an environment not available in the laboratory and, indeed, not even considered in the design criteria. The following examples selected from an unclassified document MDC-TDR-62-4 dated March 1962 are illustrative:

- It was discovered that under field conditions the electrical sensitivity of a computer caused complete failure of the computer memory.

- An analysis of the sled test data indicated a requirement for a launch site pre-run calibration to assure effective operation of the system.

- The effect of random vibration on accelerometers was found to be more detrimental than anticipated. Although this effect could not be duplicated during extensive laboratory tests, it did occur in subsequent missile flights.

- A polarity error in the readout of an accelerometer during track tests was found to have been caused by a loose connector of the type guaranteed never to come loose.

- The failure of a small structural part in a power supply whose design had previously been thoroughly laboratory tested caused failure of the entire unit.

As a result of these and many similar failures, guidance system evaluation programs now include track tests not only for the determination of error models, but also for the determination of systems reliability. Such tests for reliability are, of course, equally applicable to other systems as has been frequently demonstrated at Holloman.

In conclusion, it is believed that sufficient evidence exists to show that system reliability cannot be based solely upon component reliability which has been determined by means of laboratory tests, but must also be based upon a track test program which will subject the system to a controlled environment closely approximating the operational requirements.

DISCUSSION

J. Davis (GE): You mentioned that your facility can only take up to about 8-g deceleration. Do you have any plans, or could your facility be adopted to take much higher g levels by the use of retrorockets?

Col. Bogard: Yes, it can take much higher levels right now. As a matter of fact we can get up to 160-g. This was only on that one type of vehicle and that's because it was designed to enter the water brake at roughtly 1200 fps maximum velocity. Depending on the type of vehicle you use, there is really no limit. The only limit is the time during which you get those g's. The higher they are, of course, the shorter the time.

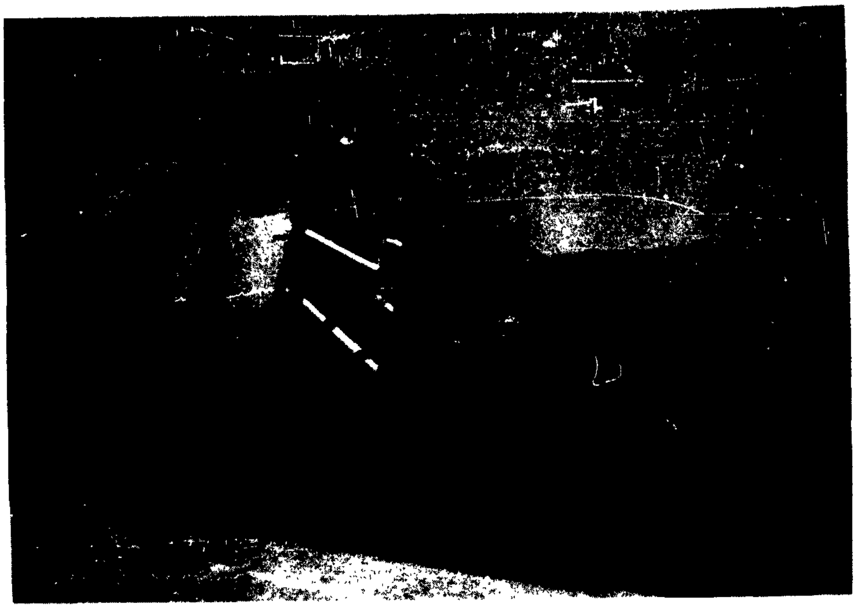

Fig. 5 - Dynasoar sled

Fig. 6 - Rain erosion damage

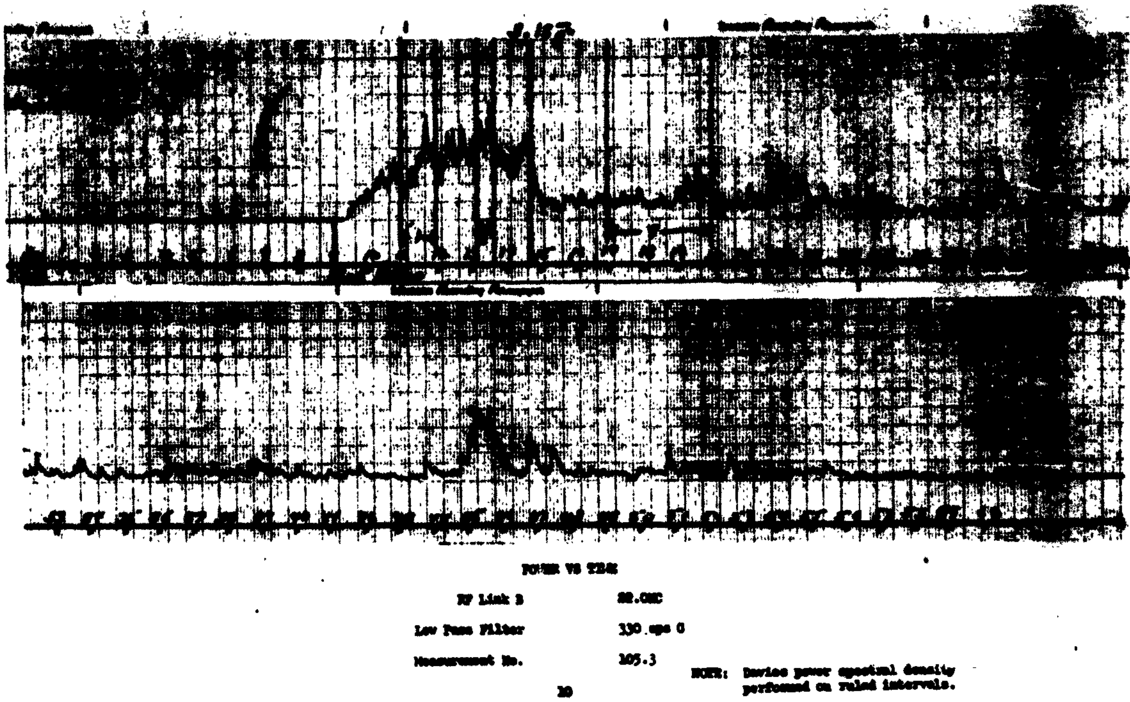

Fig. 7 - Power versus time

Mr. Schwabe (Lockheed): In regard to the sled construction, are you using air bearings or bearings with metal-to-metal contact?

Col. Bogard: The sled actually works on slippery inserts. They are just metal to metal and we have tried many, many exotic metals. We've tried air bearings on the rails, and everything else and it turns out that the best thing you can do is to use just plain old steel. On these sleds, that go roughly Mach 1.5 with about a ton payload, we find we can use the slippery inserts for three or four runs, even though we are using 20 to 30,000 feet of track. On the Pershing sled that I mentioned, that went off the end of the track at a velocity of 4000 fps, we completely wiped out the inserts on the first runs that were made. They were molybdenum inserts, very tough. They just dropped little puddles of metal all over the track. Between the second and third run, I believe it was, we ground the rails with a rail grinder and we didn't have any more trouble of that nature.

Mr. Soechting (Picatinny Arsenal): I would like to ask you, has the temperature no effect on the straightness of the track?

Col. Bogard: Yes, it has. The rails are stressed; this is one reason they are welded. They are stretched to such a tension that the ambient temperature on the rail must reach 120 degrees before the rails are in a relaxed condition. Anything less than 120 degrees Fahrenheit and the rails are held in tension just like a rubber band. This helps to hold them straight, of course.

Mr. Bond (STL): You indicated that the vibrations experienced on the sled were similar to those on a typical missile. Do you have any way of exercising control to tailor this vibration to a specific requirement?

Col. Bogard: Only to the extent that you can, and I hate to use the word, put shock mounts on the equipment. This, of course, generally leads to a long detailed development program in itself in order to find what kind of shock mounts you want to use, but there is a way, yes. On our Bosch Arma test, for instance, we are using a Brillo pad isolation on the slippers; this has reduced the vibration environment considerably. To say that we could tailor a vibration environment—no, this would be awfully difficult to do

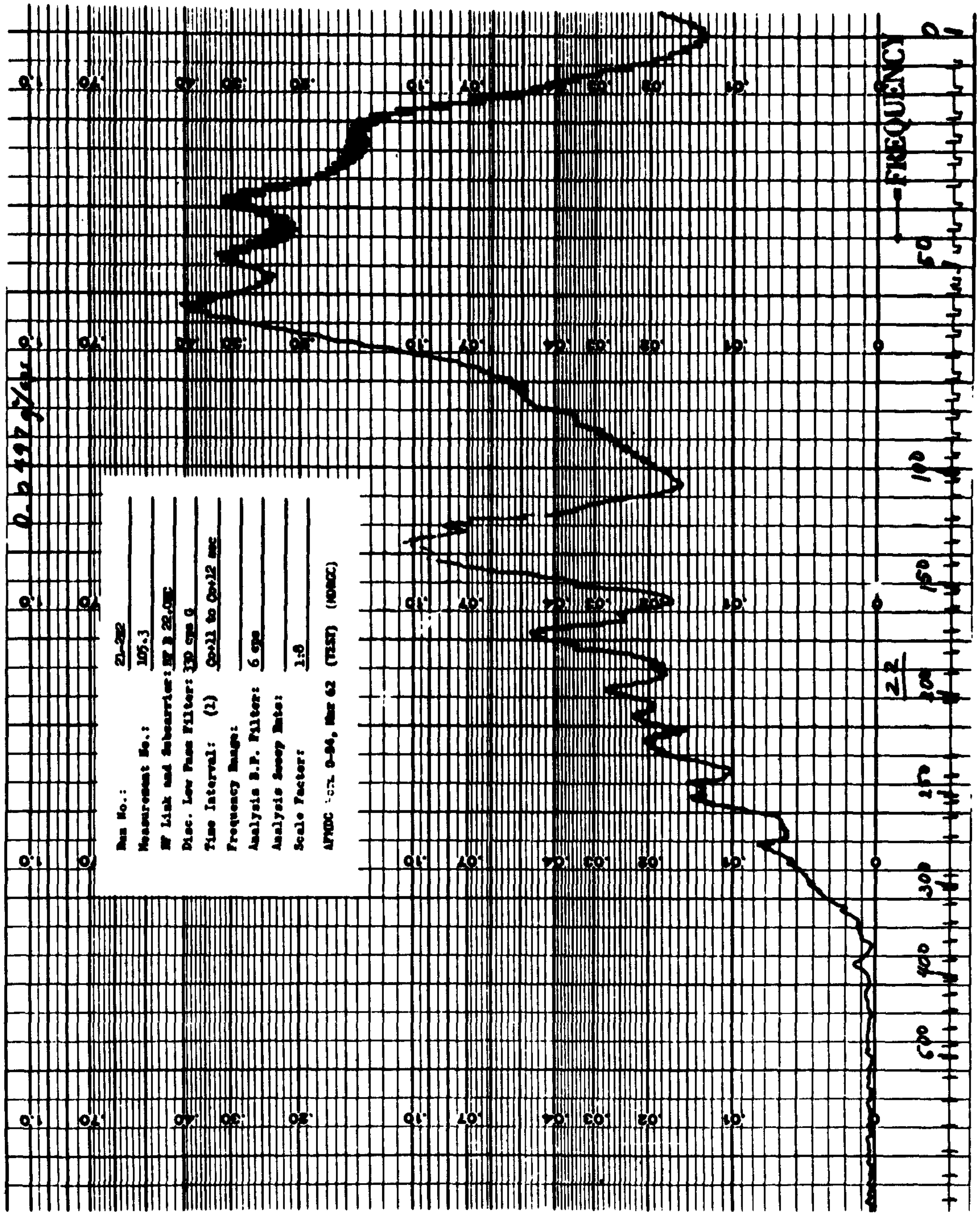

Fig. 8 - Power versus frequency

STA	M K	DIST	TC	TS	VS	SIG.	AS	TC-TS	CD
•29815.00	T	• 5205.00	•14912180	•14912183	•1407.82 •	.190 •	153.3 -	3 •	
•29802.00	T	• 5218.00	•14921414	•14921412	•1409.30 •	.217 •	142.7 •	2 •	
•29789.00	T	• 5231.00	•14930632	•14930633	•1410.39 •	.217 •	141.1 -	1 •	
•29776.00	T	• 5244.00	•14939844	•14939846	•1411.82 •	.227 •	154.6 -	2 •	
•29763.00	T	• 5257.00	•14949051	•14949048	•1413.18 •	.205 •	143.1 •	3 •	
•29750.00	T	• 5270.00	•14958243	•14958243	•1414.72 •	.248 •	136.0 •	•	
•29737.00	T	• 5283.00	•14967427	•14967430	•1415.73 •	.249 •	143.1 -	3 •	
•29724.00	T	• 5296.00	•14976607	•14976608	•1416.82 •	.264 •	149.0 -	1 •	
•29711.00	T	• 5309.00	•14985781	•14985777	•1418.57 •	.248 •	148.0 •	4 •	
•29698.00	T	• 5322.00	•14994936	•14994937	•1420.06 •	.249 •	148.0 -	1 •	
•29685.00	T D	• 5335.00	•15004085	•15004088	•1421.26 •	.236 •	129.4 -	3 •	
•29672.00	T	• 5348.00	•15013234	•15013232	•1421.99 •	.263 •	129.6 •	2 •	
•29659.00	T	• 5361.00	•15022368	•15022369	•1423.47 •	.201 •	144.6 -	1 •	
•29646.00	T	• 5374.00	•15031499	•15031497	•1425.05 •	.239 •	145.5 •	2 •	
•29633.00	T	• 5387.00	•15040614	•15040617	•1426.07 •	.222 •	148.7 -	3 •	
•29620.00	T	• 5400.00	•15049727	•15049727	•1427.57 •	.209 •	152.0 •	•	
•29607.00	T	• 5413.00	•15058832	•15058829	•1428.92 •	.210 •	143.4 •	3 •	
•29594.00	T	• 5426.00	•15067920	•15067922	•1430.48 •	.210 •	134.8 -	2 •	
•29581.00	T	• 5439.00	•15077007	•15077008	•1431.39 •	.180 •	125.1 -	1 •	
•29568.00	T	• 5452.00	•15086085	•15086086	•1432.30 •	.184 •	120.4 -	1 •	
•29555.00	T	• 5465.00	•15095160	•15095158	•1433.58 •	.144 •	142.3 •	2 •	
•29542.00	T	• 5478.00	•15104221	•15104222	•1434.92 •	.119 •	153.1 -	1 •	
•29529.00	T	• 5491.00	•15113278	•15113277	•1436.46 •	.166 •	160.3 •	1 •	
•29516.00	T	• 5504.00	•15122323	•15122322	•1437.97 •	.166 •	137.4 •	1 •	
•29503.00	T	• 5517.00	•15131359	•15131359	•1439.19 •	.152 •	128.5 •	•	
•29490.00	T	• 5530.00	•15140387	•15140390	•1440.03 •	.161 •	119.9 -	3 •	
•29477.00	T	• 5543.00	•15149415	•15149414	•1440.92 •	.182 •	144.9 •	1 •	
•29464.00	T	• 5556.00	•15158431	•15158430	•1442.56 •	.177 •	161.1 •	1 •	
•29451.00	T	• 5569.00	•15167439	•15167437	•1444.37 •	.180 •	166.4 •	2 •	
•29438.00	T	• 5582.00	•15176431	•15176433	•1445.82 •	.139 •	134.9 -	2 •	
•29425.00	T	• 5595.00	•15185422	•15185422	•1446.76 •	.135 •	117.3 •	•	
•29412.00	T	• 5608.00	•15194403	•15194404	•1447.52 •	.135 •	109.8 -	1 •	
•29399.00	T	• 5621.00	•15203383	•15203382	•1448.57 •	.116 •	135.7 •	1 •	
•29386.00	T	• 5634.00	•15212352	•15212352	•1449.85 •	.091 •	153.2 •	•	
•29373.00	T	• 5647.00	•15221315	•15221314	•1451.54 •	.101 •	166.3 •	1 •	
•29360.00	T	• 5660.00	•15230265	•15230264	•1453.13 •	.115 •	145.2 •	1 •	
•29347.00	T	• 5673.00	•15239207	•15239208	•1454.29 •	.123 •	136.0 -	1 •	
•29334.00	T	• 5686.00	•15248142	•15248143	•1455.30 •	.127 •	116.8 -	1 •	
•29321.00	T	• 5699.00	•15257075	•15257073	•1456.22 •	.173 •	126.6 •	2 •	
•29308.00	T	• 5712.00	•15265995	•15265996	•1457.53 •	.190 •	125.2 -	1 •	

•
• 210

Fig. 9 - Velocity versus time

depending on velocity obviously, and everything else. As the rail has been ground through, it has more and more come in line with the missile environment. Obviously we don't want an exact missile environment, we want it a little worse, otherwise we wouldn't know whether we were getting failure or not.

H. Sutphin (Martin Orlando): I might say that the track environment was of some concern to us. We found a very simple, very economical method of reducing the influence of the track environment. In collaboration with Lord we simply installed some rubber isolators between the slippers and our missile. It was very effective and very inexpensive.

Col. Bogard: The only problem with this type of thing, of putting in some kind of a shock isolator, is that as everybody knows it doesn't isolate at all frequencies. In one case we put in shock isolation and as it turned out it did nothing but amplify the frequency we were afraid of.

* * *

FLIGHT DYNAMIC EVALUATION OF THE BULLPUP B MISSILE

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Martin Company, Orlando, Florida Division
Martin Marietta Corporation

The purpose of this paper is to summarize the available flight data and the analyses which were made to establish the maximum BULLPUP B component shock, vibration, and steady acceleration requirements for specification comparisons and design purposes.

INTRODUCTION

The Navy's BULLPUP B is an air-to-surface command-guided missile which may be launched from carrier and land-based attack and fighter aircraft. The missile body comprises three sections. The nose section houses the guidance and control system. The center section carries the warhead, and the aft section is the propulsion unit. BULLPUP B is an outgrowth of the smaller BULLPUP A missile now operational with the Navy as the ASM-N-7a and with the Air Force as the GAM-83A. Both missiles use the same nose section. The earlier experimental "XB" version was flown with a solid propellant engine. The developmental "YB" version uses a prepackaged liquid propellent unit, as originally proposed. The BULLPUP B is now in the evaluation phase of its development and is expected to be operational in 1963.

The simplicity of the BULLPUP system is apparent in that only four component areas are considered critical to missile operation from a dynamic standpoint. They are the guidance command receiver, the roll reference gyro, the controls package, and the warhead fuze.

At the outset of the flight test program, it was decided to obtain a large sample of data from two component areas (the guidance command receiver and the roll reference gyro, as shown in Fig. 1) rather than a necessarily minimal amount of information from all four positions. Therefore, this paper describes the flight environments of the receiver and gyro areas only.

The BULLPUP B nose and center sections may be used in conjunction with either solid propellant or prepackaged liquid propellant propulsion units; therefore, major emphasis was placed upon definition of the maximum environmental requirements for either configuration as the fundamental environmental description.

FLIGHT ENVIRONMENT

The flight environment consists of four distinct phases: captive flight on the launching aircraft, ejection launch, boost or powered flight, and glide or power-off flight.

Captive Flight

The captive flight phase of missile usage encompasses the period of time from missile installation on the aircraft until launch (ejection) or removal. Because of telemetry battery life limitations, the captive flight analysis was confined to a portion of the unit operating time prior to launch (approximately 3 seconds).

No appreciable shocks were noted in the captive period analyzed. Outside this time span, however, component shocks are anticipated during catapult takeoffs and arrested landings. Data available from previous BULLPUP A tests indicate that these shock levels should be appreciably below specification test amplitudes.

Representative captive flight vibration data in the form of power spectral density graphs are presented in Figs. 2 and 3.

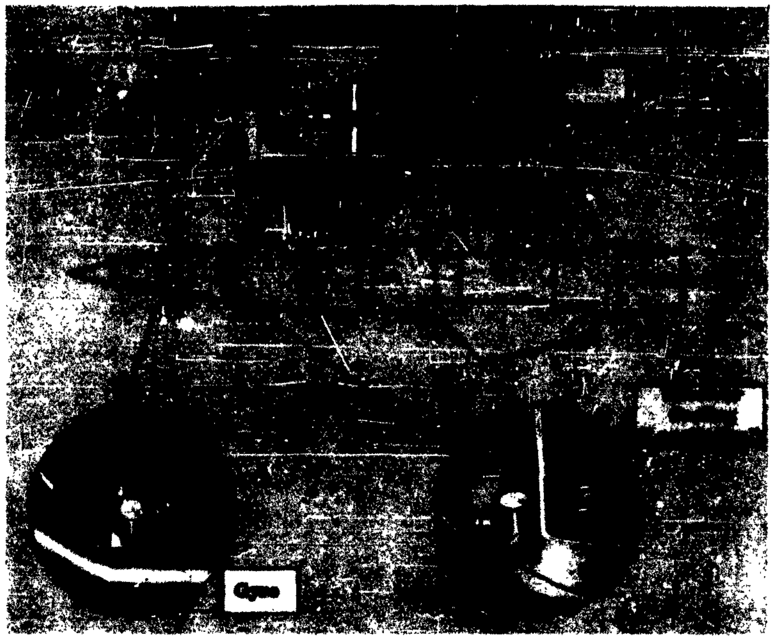

Fig. 1 - Accelerometer locations

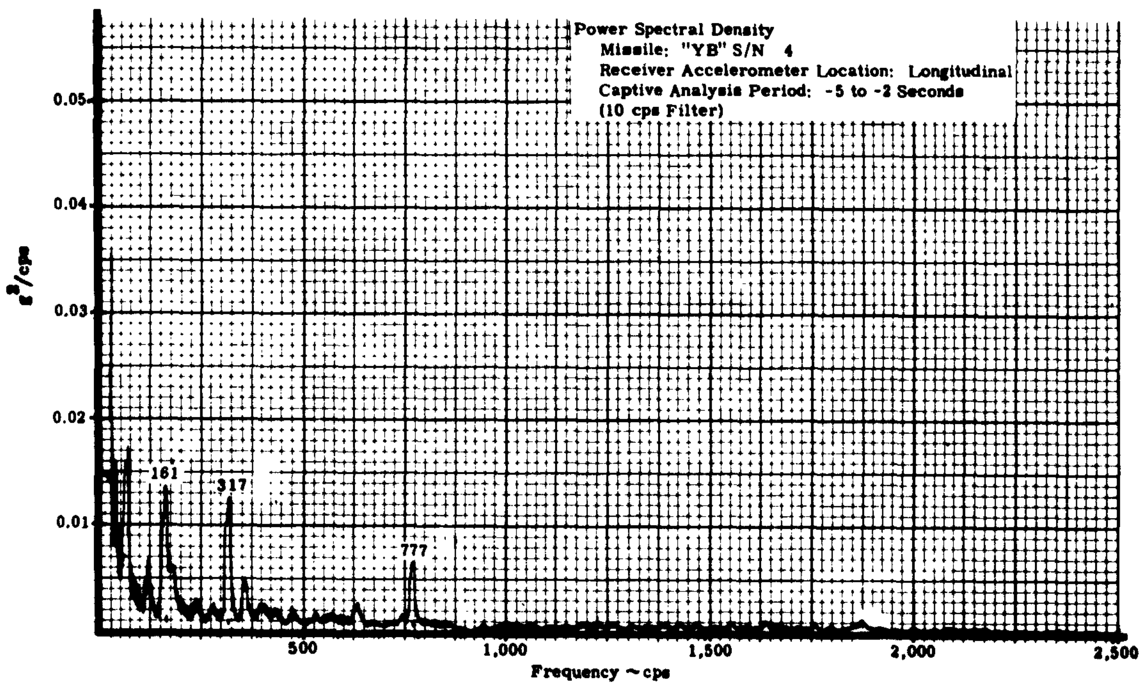

Fig. 2 - Power spectral density, receiver longitudinal, "YB-4"

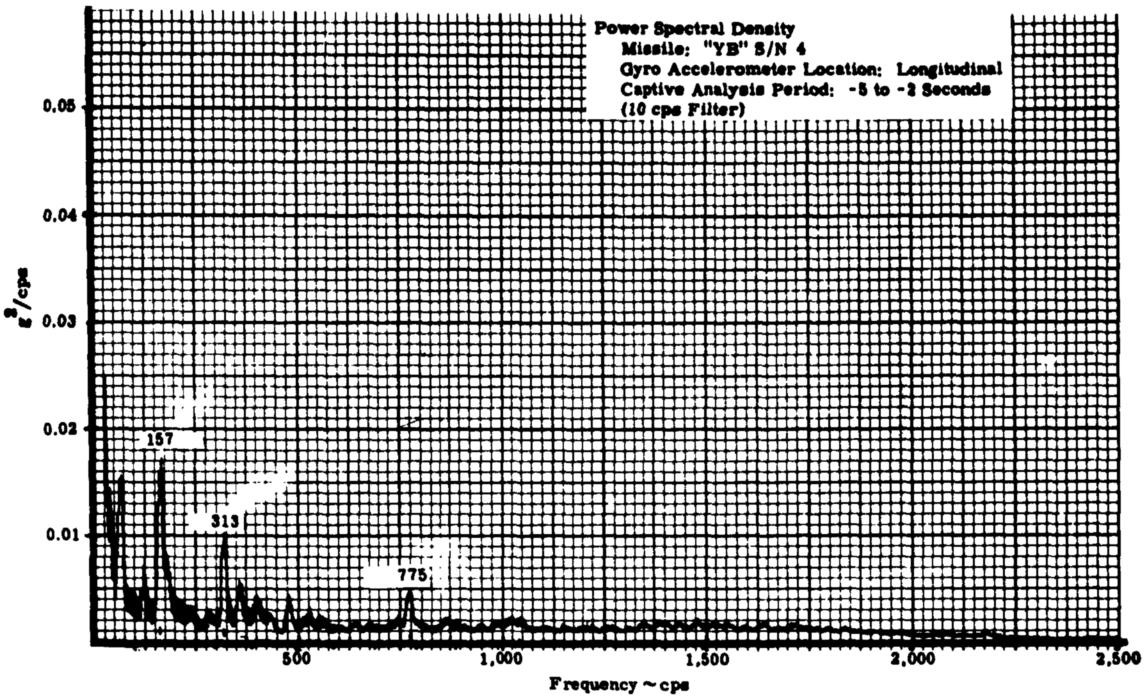

Fig. 3 - Power spectral density, gyro longitudinal, "YB-4"

Since the scope of the planned captive flight test did not encompass a representative sample of aircraft maneuvers, it was not possible to analyze completely the accelerations in this period. The catapult takeoff and arrested landing design acceleration levels are shown in Table 1.

Ejection Launch

The BULLPUP B is ejection-launched from either an Aero 20A or Aero 7A bomb rack. The separation impulse (normal to aircraft and missile roll axes) is provided by an ejector foot which is propelled by a pyrotechnic cartridge charge.

With various ejector-foot-to-missile clearances, ejection causes component shocks of differing magnitudes. Tables 2 and 3 illustrate ground and flight test results from programs conducted to determine these shock characteristics.

The missile normal axis acceleration during ejection is approximately 20 percent of the specification level.

Powered Flight

For the purposes of analysis, the powered flight stage for both the solid and liquid propellant units was subdivided as follows:

- Ignition Phase (shock) - The first 0.250-second period after the ignition signal.

- Running Phase (vibration) - That portion of operation remaining after the ignition phase.

Shock analysis results for the "XB" (solid propellant unit) and "YB" (prepackaged liquid propellant unit) configurations are presented in Tables 4 and 5. It is apparent from this data that the component shock requirements are greater for the prepackaged liquid propellant unit system than for the solid propellant configuration. In addition, it is evident (Fig. 4) that the liquid propellant unit ignition shock magnitudes are dependent upon launching aircraft attitude. Previous ground test results had shown that a maximum impact condition does occur with the liquid propellant unit at approximately the same unit attitude recorded in flight.

Typical vibration intensities for both configurations are represented by power spectral

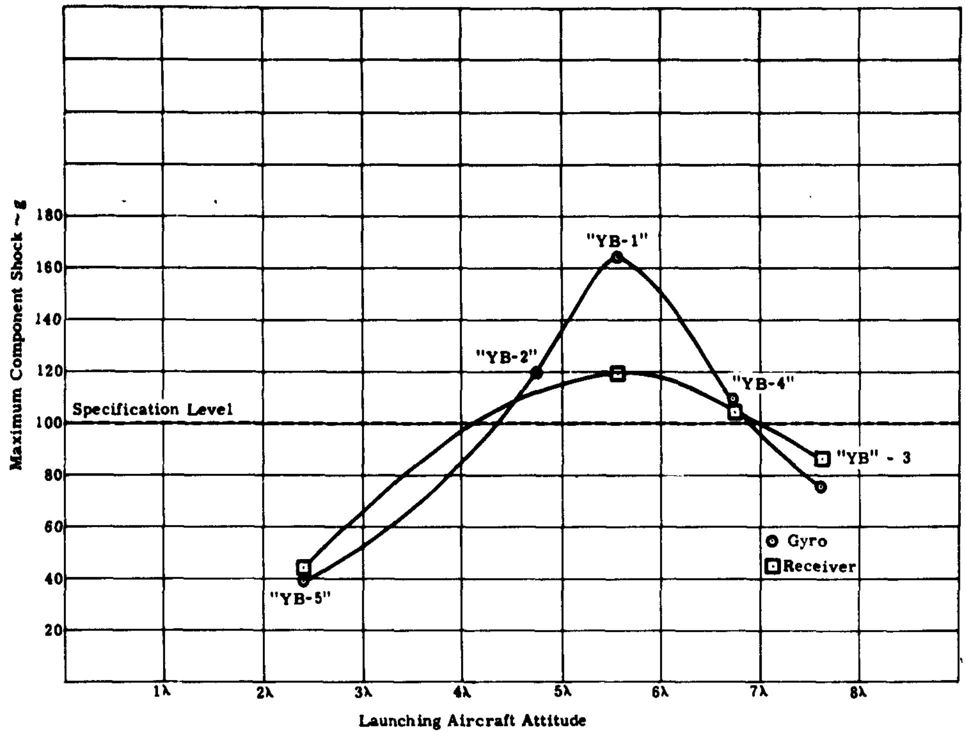

Fig. 4 - Aircraft attitude versus component shock

density graphs (Figs. 5 through 8). The "XB" results indicate a relatively low level "white noise" vibration with no definite areas of maximum response. The "YB" analyses show generally a relatively low level "white noise" vibration with maximum responses occurring in the frequency band 200-250 cps, depending upon the flight examined. This maximum response occurs consistently (within the defined band) at both accelerometer locations during all "YB" flights and, therefore, has been designated as the primary longitudinal response mode of the "YB" missile.

Also, analysis of the one "YB" flight (YB-2) with normal axis instrumentation at the receiver, shows peaked response at 221 cps and the highest RMS vibration level (Table 6), indicating that considerable system crosstalk (i.e., excitation in a plane normal to the primary vibration axis) is present. Amplitude distribution analyses performed on both systems (Figs. 9 through 12) show that the "XB" amplitude distribution is very nearly Gaussian while the "YB" distributions are considerably different from the normal case. The "YB" results illustrate that application of the commonly used Gaussian distribution for component testing purposes will be conservative. That is, the probability of Gaussian test amplitude occurrences above the RMS level is approximately 10 percent greater than the corresponding average flight values.

The longitudinal axis results (Table 7) show that the maximum recorded "XB" and "YB" peak accelerations are 92.5 and 84.0 percent of the specification level, respectively.

Power-Off Flight

Power-off flight, the last phase of missile operation, covers the period from propulsion unit burnout until impact.

Oscillograph records of controlled flights show intermittent shocks indicative of control surface operation during guidance commands. The gyro experiences the highest longitudinal responses of 60 and 75 g on particular isolated occasions. Generally, however, the responses are below 40 g, if at all measurable, and always less than 0.001-second duration.

Typical vibration intensities during power-off flight are represented by power spectral density graphs (Figs. 13 and 14).

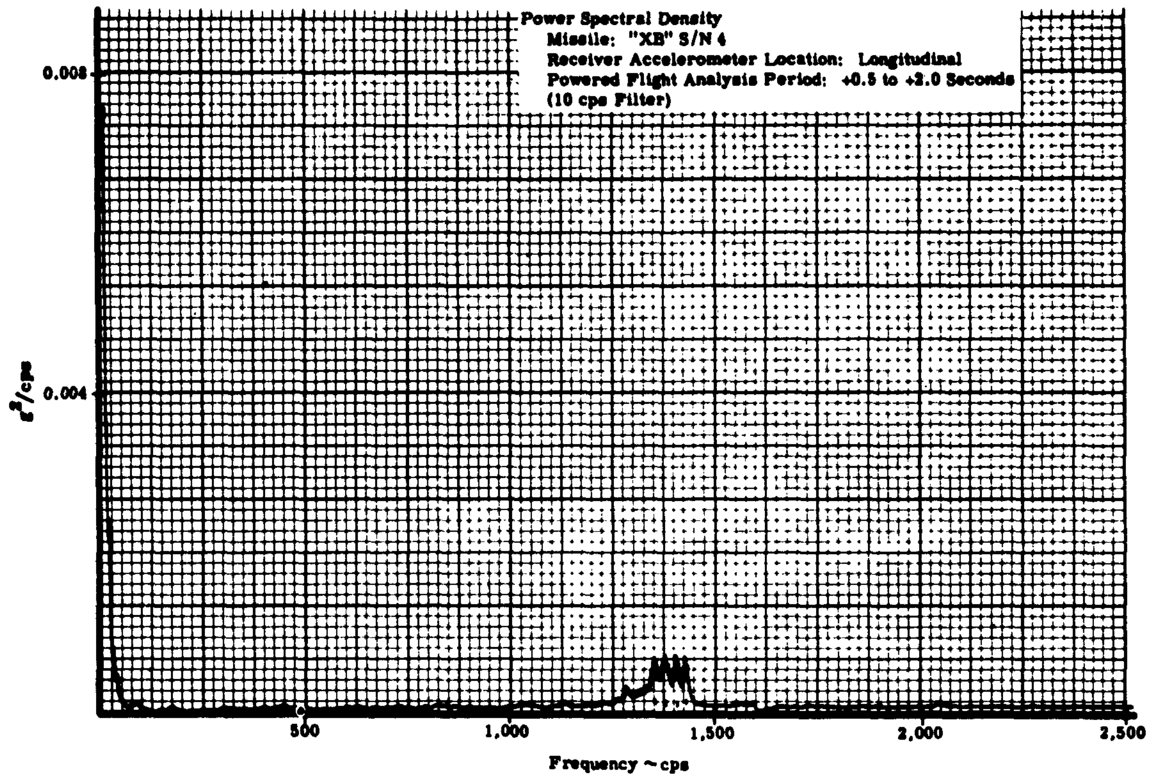

Fig. 5 - Power spectral density, receiver longitudinal, "XB-4"

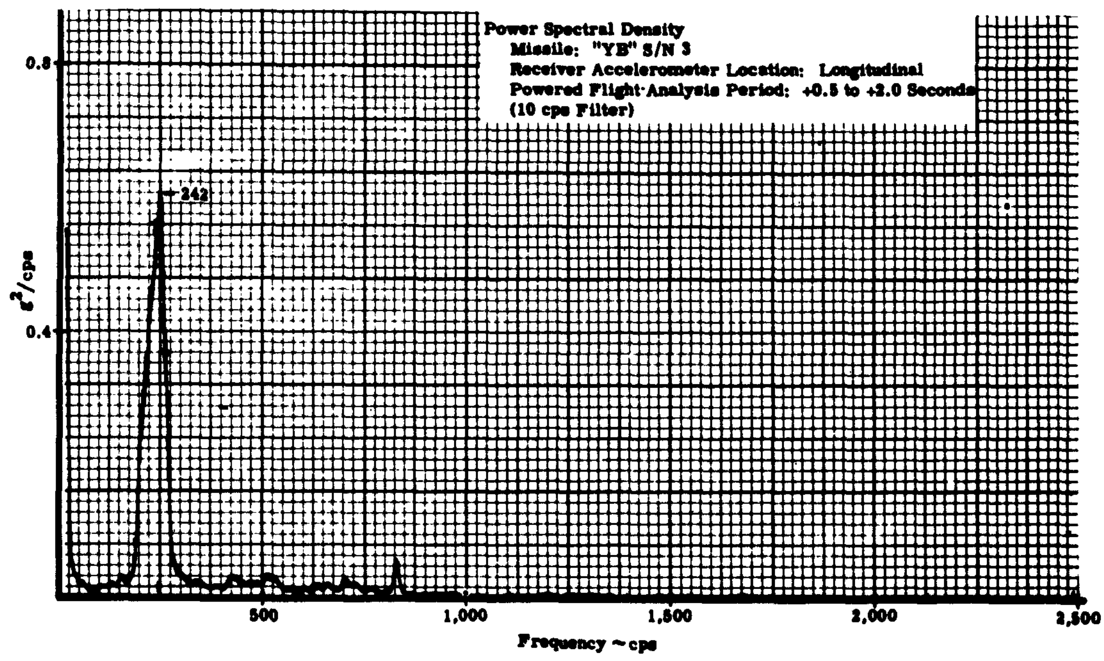

Fig. 6 - Power spectral density, receiver longitudinal, "YB-3"

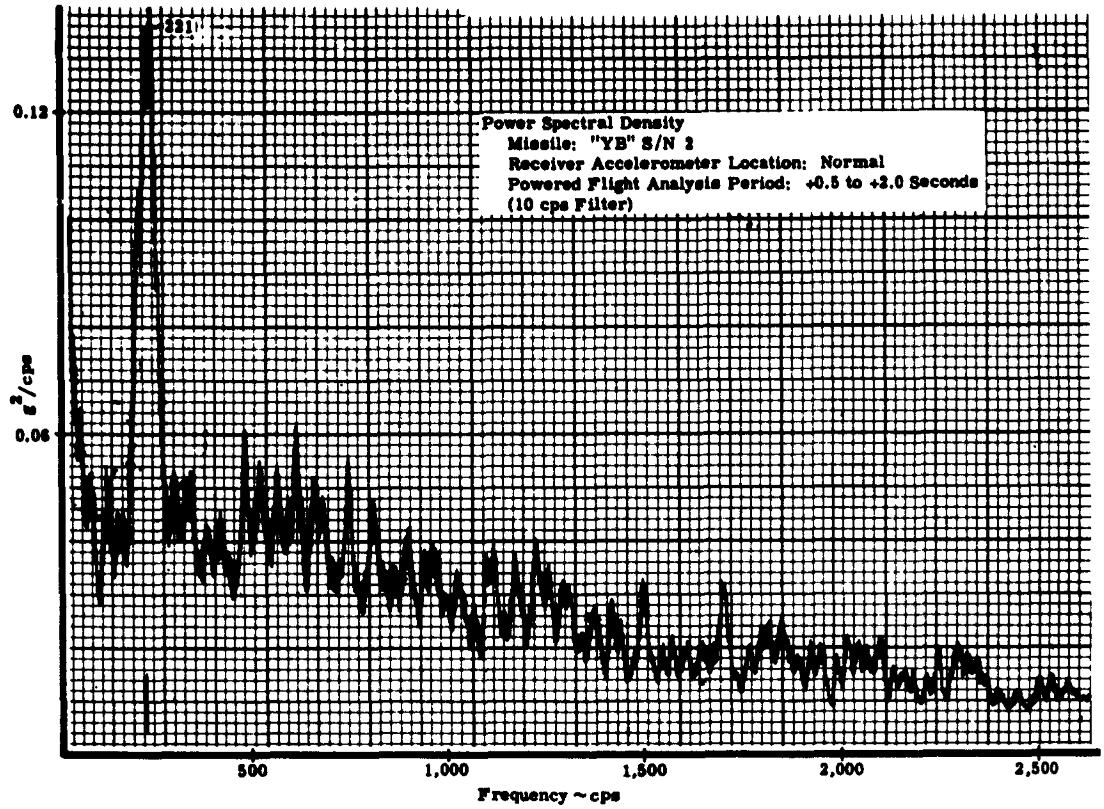

Fig. 7 - Power spectral density, receiver normal, "YB-2"

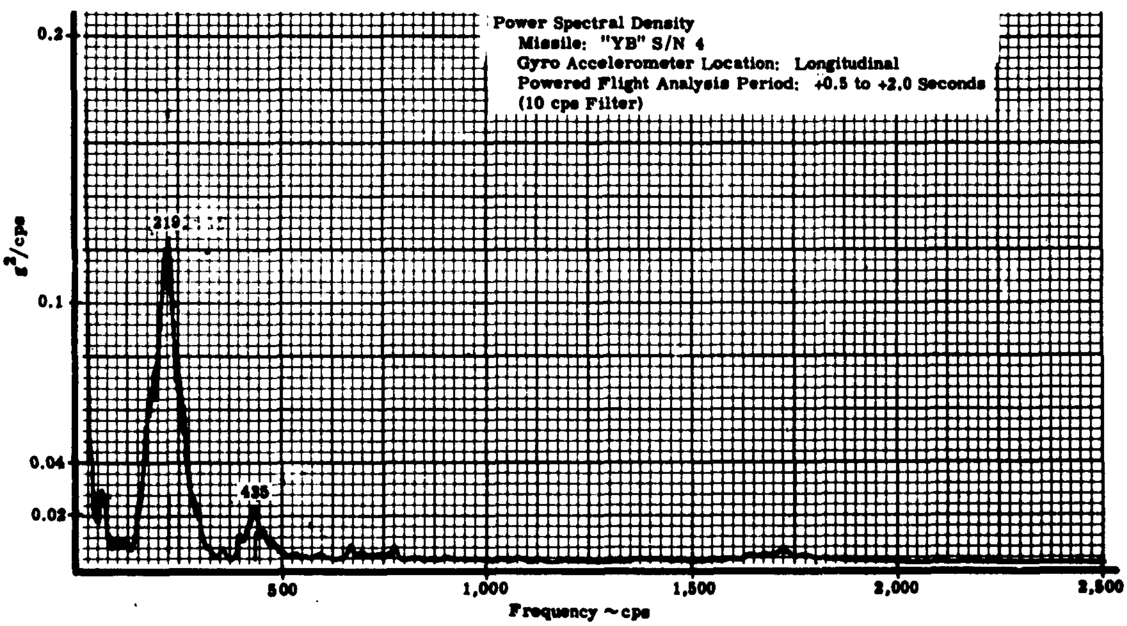

Fig. 8 - Power spectral density, gyro longitudinal, "YB-4"

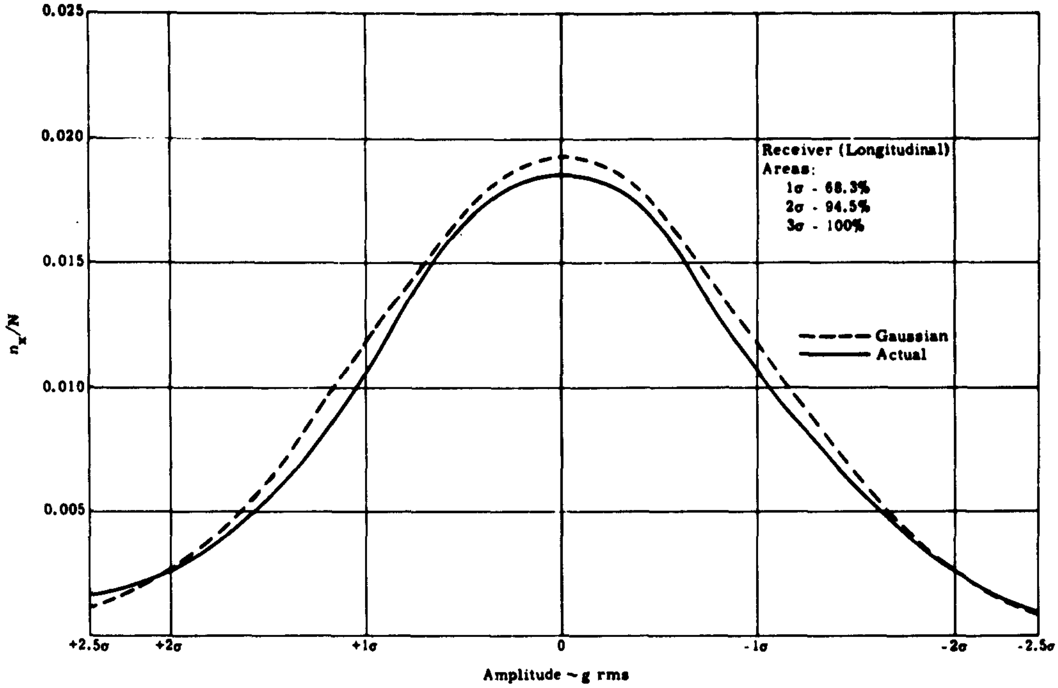

Fig. 9 - Amplitude distribution, receiver longitudinal, "XB-1"

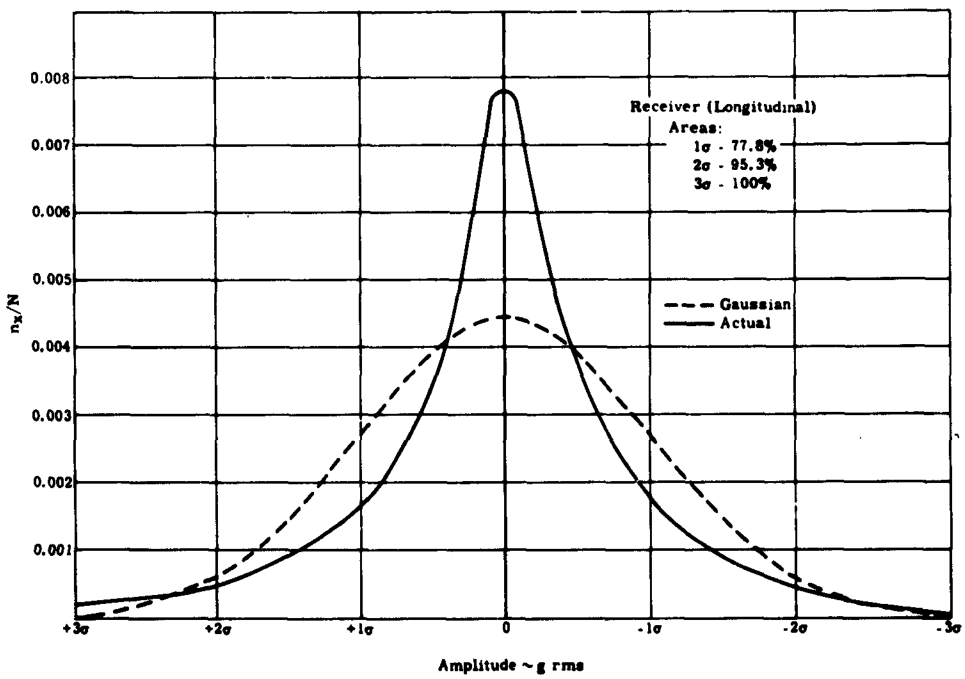

Fig. 10 - Amplitude distribution, receiver longitudinal, "YB-4"

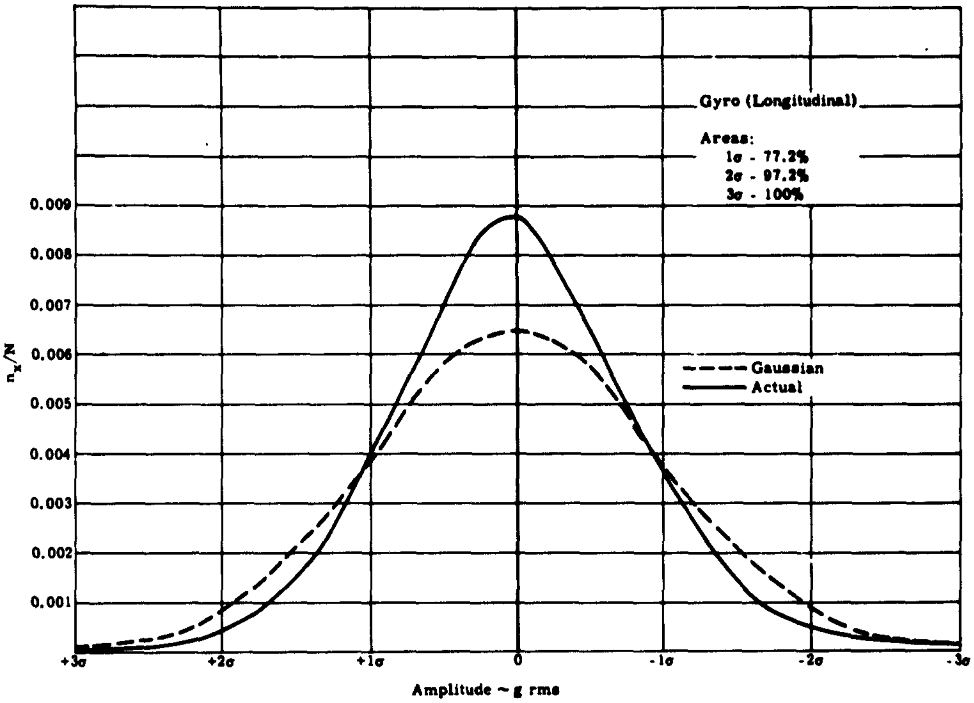

Fig. 11 - Amplitude distribution, gyro longitudinal, "YB-2"

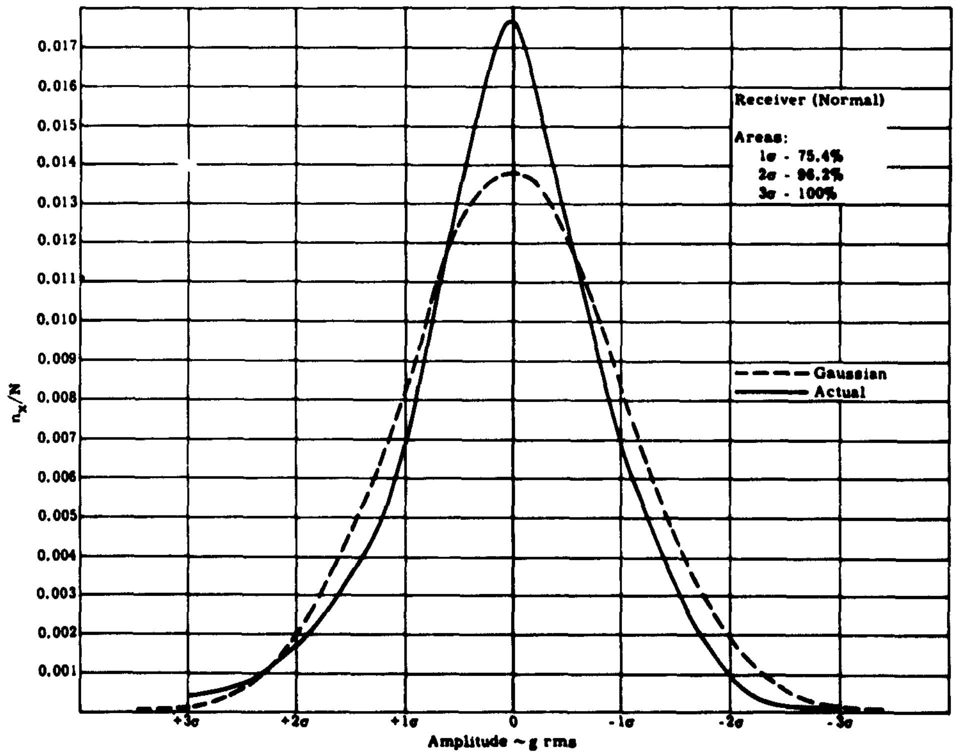

Fig. 12 - Amplitude distribution, receiver normal, "YB-2"

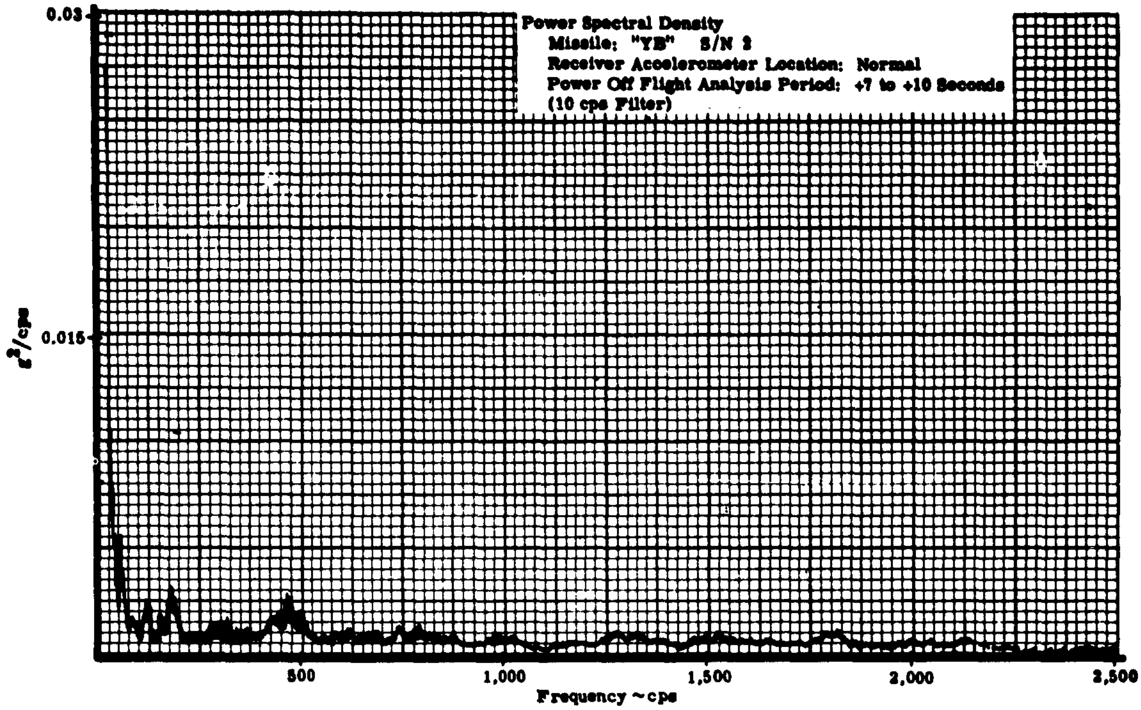

Fig. 13 - Power spectral density, receiver normal, "YB-2"

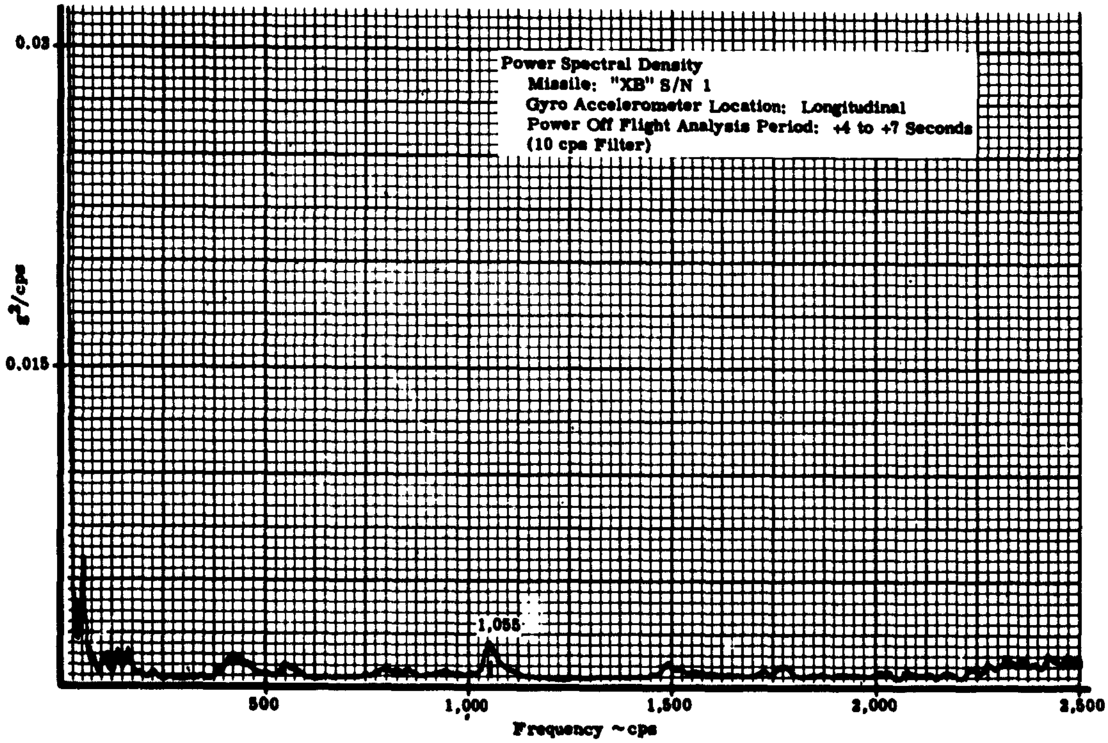

Fig. 14 - Power spectral density, gyro longitudinal, "XB-1"

The missile, in responding to commands, experiences a rapid change in angle-of-attack. Accordingly, the maximum accelerations during power-off flight are attributed to missile command responses. Because the missile is continually rolling during flight, effective normal and transverse axis component accelerations are vector additions of pitch and yaw accelerations. The maximum recorded total acceleration during power-off-flight of both configurations is 22 percent of the specification steady acceleration requirement.

SUMMARY OF RESULTS AND CONCLUSIONS

The data and the analysis results establish the powered flight phase of missile operation as the fundamental (maximum) BULLPUP B dynamic environment with but two exceptions— the maximum normal and transverse acceleration conditions occur during captive flight and power-off flight, respectively.

BULLPUP B components are qualified dynamically to the following requirements:

Shock

Longitudinal axis (both directions)......100 g (0.005 sec)
Normal to longitudinal axis .... 15 g (0.011 sec)

Vibration (All three principal axes)

10-1200 cps at ± 5 g
1200-3000 cps at ± 9 g

Acceleration

Classified

The following conclusions are drawn from a consideration of the data and the analyses.

Shock

Longitudinal shocks at the gyro and receiver above the qualification amplitude were recorded at liquid-propellant engine ignition on several occasions during the "YB" flight tests (Table 5). Receiver normal-axis responses in excess of the qualification amplitude were recorded during both flight test (for the one missile instrumented in the normal axis) and the ground ejection test (Tables 2 and 5). With the exception of some ground test shocks, the pulse durations in all cases were less than the qualification values.

The longitudinal shock qualification amplitudes for the gyro and receiver will apparently be exceeded by shorter-duration acceleration pulses during "YB" flights made at aircraft attitudes in the range shown in Fig. 4.

These results indicate that BULLPUP B guidance components will experience shocks of greater amplitude, but shorter duration, than those required by present specifications.

Vibration

Presently, there is no proven method of directly comparing random and sinusoidal vibration criteria. In this particular case, though, several useful comparisons may be made. The present qualification spectrum is flat in the frequency bands 10-1200 and 1200-3000 cps in all three principal test axes. Typical longitudinal "YB" results (Figs. 7 and 8) show generally a low level flat response in the bands 10-200 and 250-2100 cps (2100 cps is the maximum telemeter playback response) with relatively high responses in the band 200-250 cps. Based on these spectrum analyses and on approximate oscillograph-measured component magnitudes in the band 200-250 cps, the longitudinal test spectrum is considered adequate in the frequency bands 10-200 and 250-2100 cps and inadequate in the band 200-250 cps. This conclusion is also applicable to the normal and transverse axes.

BULLPUP B components are qualified to sinusoidal vibration requirements. It is evident, however, that available flight acceleration time histories display "randomness" in both frequency and amplitude, and can only be analyzed adequately by applying random analysis techniques (power spectral density and amplitude distributions). Random vibration test techniques, therefore, are considered a more accurate environmental duplication for BULLPUP B component qualification purposes than sinusoidal techniques and should be applied in future component qualification.

The flight analyses also illustrate that application of the maximum flight vibration environment is entirely within the scope of present random testing facilities.

Acceleration

The qualification level for acceleration is adequate in all three major test axes. The recorded levels in the longitudinal axis are within 7.5 percent of the specification values while the maximum normal and transverse axis accelerations are less than half the qualification level.

TABLE 1
Maximum Anticipated Catapult and Arrested Landing Accelerations (Wing Station Store)

Acceleration Cause	Longitudinal (roll axis) (% of Specification Level)	Normal (yaw axis) (% of Specification Level)	Transverse (pitch axis) (% of Specification Level)
Arrested Landing	30	27	5
Catapult	30	10	5

TABLE 2
"YB" Ground Ejection Launch Shocks

Accelerometer Location and Orientation	Aero 20 A Ejector Foot Clearance							
	0		3/16 (inch)		1/2 (inch)		1 (inch)	
	Max. Accel. (g)	Duration (sec)	Max. Accel. (g)	Duration (sec)	Max. Accel. (g)	Duration (sec)	Max. Accel. (g)	Duration (sec)
Receiver Casting (Longitudinal)	19.0	0.0011	19.0	0.0012	22.0	0.0013	19.0	0.0011
Receiver Casting (Vertical)	19.0	0.010	15.0	0.010	20.0 14.0	0.0020 0.010	28.0 23.0	0.0020 0.010
Gyro (Longitudinal)	19.0	0.0017	11.0	0.0025	21.0	0.0014	21.0	0.0015

TABLE 3
"YB" Flight Test Ejection Shocks

Missile Number	Accelerometer Location			
	Receiver		Gyro	
	Max. Accel. (g)	Duration (sec)	Max. Accel. (g)	Duration (sec)
1	16.0	0.002	19.0	0.0015
2	15.0	0.004	10.0	0.0012
3	Low	--	Low	--
4	20.0	0.0015	10.0	0.0015
5	Low	--	Low	--

TABLE 4
Maximum "XB" Ignition Shock Summary

Accelerometer Location and Orientation	Missile Number									
	1		2		3		4		5	
	Accel. (g)	Dura. (sec)	Accel. (g)	Dura. (sec)	Accel. (g)	Dura. (sec)	Accel. (g)	Dura. (sec)	Accel. (g)	Dura. (sec)
Gyro (Longitudinal)	12	0.001	5.0	0.001	5.0	0.001	13	0.001	20	0.001
Receiver (Longitudinal)	10	0.001	30	0.001	5.0	0.001	25	0.001	12	0.001

TABLE 5
Maximum "YB" Ignition Shock Summary

Accelerometer Location and Orientation	Missile Number									
	1		2		3		4		5	
	Accel. (g)	Dura. (sec)	Accel. (g)	Dura. (sec)	Accel. (g)	Dura. (sec)	Accel. (g)	Dura. (sec)	Accel. (g)	Dura. (sec)
Gyro (Longitudinal)	164 129	0.0013 0.0013	119	0.0013	75	0.0015	109 86	0.0018 0.0017	39 39	0.0014 0.0015
Receiver (Longitudinal)	119 111	0.0022 0.0020			86 58	0.0012 0.0020	103 101	0.0016 0.0014	44 39	0.0018 0.0017
Receiver (Normal)			47	0.0013						

TABLE 6
Summarized Power Spectral Density
RMS Levels ("YB"; "XB")

Missile and Location	RMS Level (Captive Flight)	RMS Level (Motor Burning)	RMS Level (Power-Off Flight)	Accelerometer Orientation
"XB" - 1				
Receiver		1.18	1.44	Longitudinal
Gyro	1.205	N. A.	1.27	Longitudinal
"XB" - 4				
Receiver	0.340	0.400	0.358	Longitudinal
Gyro	0.461	N. A.	0.628	Longitudinal
"YB" - 1				
Receiver	1.676	6.725	1.915	Longitudinal
Gyro	1.775	5.220	2.875	Longitudinal
"YB" - 2				
Receiver	1.292	8.20	1.241	Normal
Gyro	1.565	N. A.	1.612	Longitudinal
"YB" - 3				
Receiver	2.27	7.40	2.765	Longitudinal
Gyro	1.565	--	--	Longitudinal
"YB" - 4				
Receiver	1.697	5.870	2.13	Longitudinal
Gyro	1.675	4.122	2.23	Longitudinal
"YB" - 5				
Receiver	2.324	3.740	2.320	Longitudinal
Gyro	2.090	3.49	2.425	Longitudinal

TABLE 7
Summary of Peak Motor Boost Acceleration (Longitudinal)

"XB" Missile Number	Acceleration (% of Specification Level)	"YB" Missile Number	Acceleration (% of Specification Level)
"XB" - 1	71.8	"YB" - 1	63.5
"XB" - 2	72.8	"YB" - 2	66.5
"XB" - 3	--	"YB" - 3	63.5
"XB" - 4	79.5	"YB" - 4	84.2
"XB" - 5	--	"YB" - 5	63.5
"XB" - 6	75.2	"YB" - 6	63.1
"XB" - 7	69.5	"YB" - 7	67.5
"XB" - 8	79.2	"YB" - 8	64.5
"XB" - 9	84.2		
"XB" - 10	66.8		
"XB" - 11	92.5		

DISCUSSION

Mr. Davis (GE): You said that you measured a shock at a component location due to a shock or transient input to the system. Now isn't it really true that what you measured was decaying vibration rather than a shock . . . the frequency of that vibration probably being related to the item you are talking about. The thing I am concerned about is that if the component resonant frequency is near that vibration frequency then a shock test may really be very unconservative because with a decaying vibration you might be getting amplifications of four or five or six, where with a shock test, you might only be getting two.

Mr. Hodge: You mean with a conventional half sine method as opposed to duplicating the whole wave form:

Mr. Davis: Yes.

Mr. Hodge: You are right.

Mr. Davis: The question is really, is a shock test a valid test for a component for qualification when it results from a system condition?

Mr. Hodge: Well, that is what the component saw and as close as we can come to it right now, is a half sine test.

Mr. Davis: Well, one of the things we are thinking about in certain shock applications is the necessity for actually doing a high-g-level vibration test on a component to simulate the transient vibrations . . . if we feel this would be important.

Mr. Schwabe (Lockheed): Am I correct in assuming that your psd plots, were corrected for noise, frequency distortion, and so on?

Mr. Hodge: Yes, they are corrected.

Mr. Schwabe: Did you make any attempt to statistically combine any of the psd plots from the same combinations to come up with a statistical mean and extreme?

Mr. Hodge: You mean between flights?

Mr. Schwabe: Yes. You have several missiles and, assuming you have the psd plots at any one station, you have perhaps about six or eight samples depending on how many missiles you flew.

Mr. Hodge: We felt that the sample was too small. We are going to supplement this with some flights later on and then we would like to attempt to do it.

* * *

EFFECTS OF WEIGHT ON THE FREQUENCY AND AMPLITUDE OF VIBRATION TEST ITEMS*

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Environmental Division
Deputy for Test and Support
Aeronautical Systems Division, AFSC

Presented in this paper are the results of a survey which was made to investigate the effects of weights, in an in-flight vibration environment, on the dynamic characteristics of equipment.

INTRODUCTION

It has often been said that the dynamic characteristics of items measured in flight will change appreciably as the weight of the items is increased, but little experimental data have been presented in support of this contention. In the course of a comprehensive data acquisition program, the Aeronautical Systems Division, USAF, conducted a survey (mass-weight survey) to investigate the effects of weight on the dynamic response of items in an in-flight vibration environment... notably on the upper frequency cutoff (the frequency above which no data exceeds ±0.2 g) and the low-frequency amplitude variation. Hitherto consideration of these dynamic characteristics has been restricted to the specifications for transportation of special equipment items; however, with the advent of a new environmental standard, study of these characteristics has been extended to include operating equipment.

The results of this survey were submitted as raw data to the requesting agency for the preparation of a frequency-versus-weight chart to be incorporated into the Ground Support Equipment Specification. This paper has been prepared to give a wider distribution to the results of the mass-weight study. The chart is applicable only to the transportation procedures of the specification.

TEST PROCEDURE

For the mass-weight survey, an RB-50 aircraft was instrumented with 33 vibration pickups, installed at 11 points of interest, to sense the vibration along the 3 major axes. Eighteen of the vibration pickups were mounted directly on the test items; the other fifteen were positioned on the structures adjacent to the test items.

The locations of the vibration pickups in the aircraft are shown in Fig. 1 and described in Table 1. Mass-weight of 40, 150, and 400 pounds were alternated at the same location in the nose section of the aircraft to avoid data differences due to positioning. Figure 2 shows one of the mass-weight assemblies positioned in the aircraft.

A total of 14 test flights were conducted and measurements were taken during all normal flight conditions such as taxi, ground runup, takeoff, straight and level flight (at various altitudes, airspeeds, and power settings), descent, and landing. Approximately 14,000 data points were obtained from the pickups at the 11 locations during the 14 test flights. The reels of recorded data were edited in the laboratory, and each sample (approximately 5 seconds in length) was spliced into an endless loop. These loops were then placed on a Davies Model 502 tape playback system and a narrow bandwidth (10 cps) analysis from 5 to 500 cps was conducted simultaneously on six channels by using a Davies Model 510 heterodyne analyzer. The analyzed data were recorded on six modified brown strip chart recorders in the form of a continuous spectrum of frequency (cps) versus transducer voltage (rms). The data points of interest were then extracted from the strip chart recordings, tabulated, and punched into

*This paper was not presented at the Symposium.

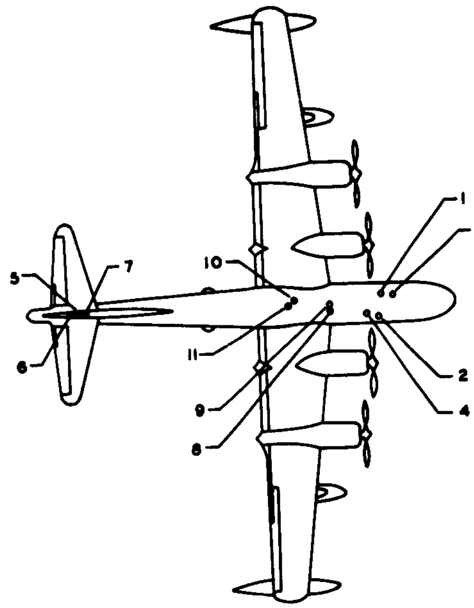

Fig. 1. Vibration pickup locations on RB-50 aircraft

Fig. 2. Mass weights used for test (B-50 vibration survey)

TABLE 1
Vibration Pickups

Pickup No.	Location
1	On left side of mass-weight assembly
2	On right side of mass-weight assembly
3	On structure forward of mass-weight assembly
4	On structure aft of mass-weight assembly
5	On 40-pound camera in tail compartment (shockmounted)
6	On 100-pound camera in tail compartment (shockmounted)
7	On structure in tail compartment
8	On 500-pound bomb on forward bomb rack
9	On structure of forward bomb rack
10	On 1000-pound bomb on aft bomb rack
11	On structure of aft bomb rack

IBM cards. Corresponding decks of "master cards," which contain detailed descriptive information concérning pickup locations, flight conditions, and source and order of the vibration, were also prepared. The extracted data and the appropriate descriptive information were processed by means of an ERA 1103 computer. Both the vibratory double amplitude (expressed in inches) and the acceleration (expressed in "g" units) appear in the completed data card. The data were then sorted into the desired order and graphed by an automatic plotter having IBM card input capabilities.

TEST RESULTS

This report discusses only the test envelopes derived from the experimental data, since the envelopes suffice for the purpose of this survey and simplify the data presentation. The frequency and amplitude responses of each test item and its adjacent structure to the vibration environment are plotted together for comparison. The pairs of plots are presented in Figs. 3 through 9 in the order of increasing test item weight to indicate the trend of the weight effects. Each log-log curve, of frequency in cycles per second and double amplitude in inches, depicts the cutoff frequency of the measured item or structure.

Figures 3 through 9 reveal the decrease in magnitude of the upper frequency cutoffs as the weights of the test items increase. Table 2 lists the cutoff frequency of each test item. A plot of these frequency cutoff magnitudes versus the weights indicates a smooth curve. This curve along with one based on theoretical data is presented in Fig. 10. Although the sampling might not be sufficient, it appears that the weight of any test item would have an upper frequency cutoff corresponding to the ordinate value of the curve based on the measured data. Further, given the same test item weight regardless of the nature of the item, the manner

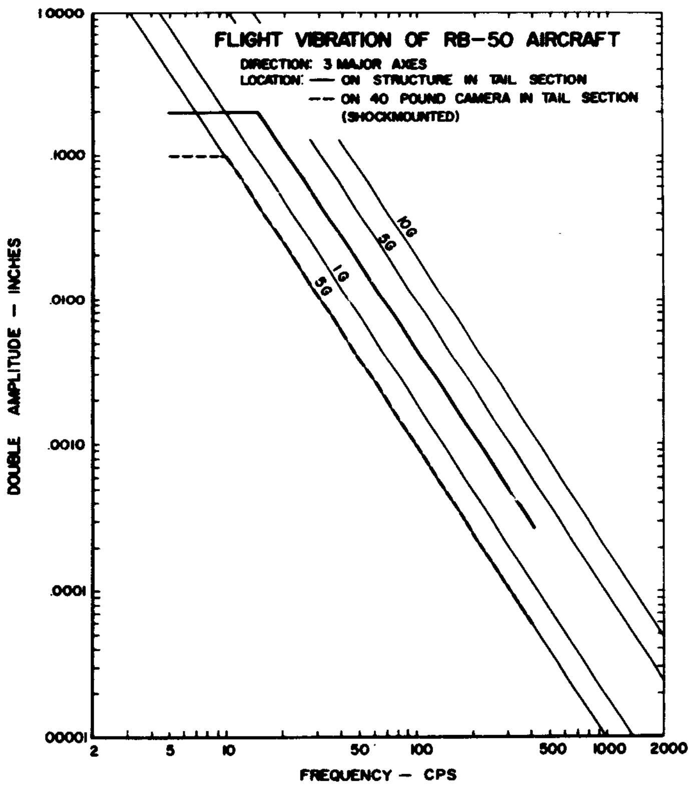

Figure 3

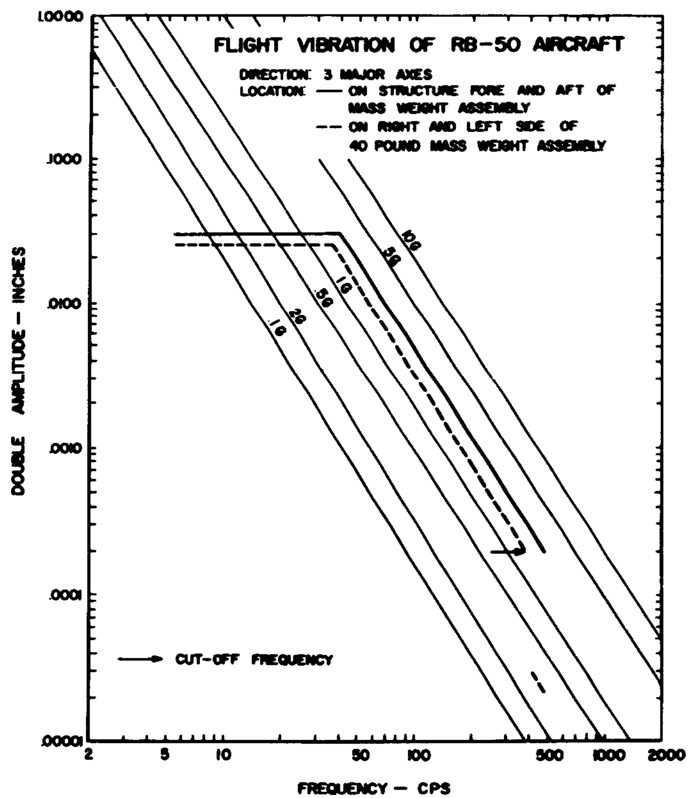

Figure 4

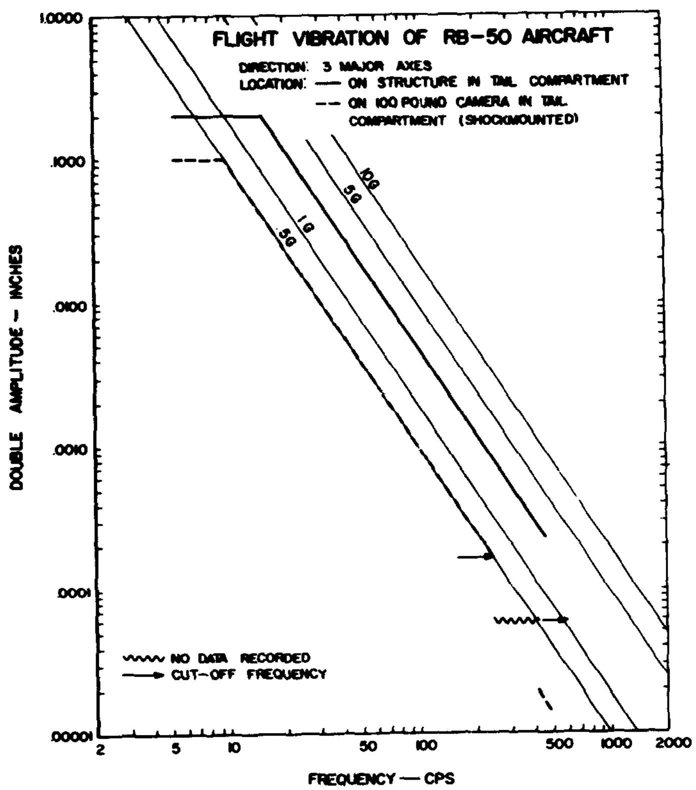

Figure 5

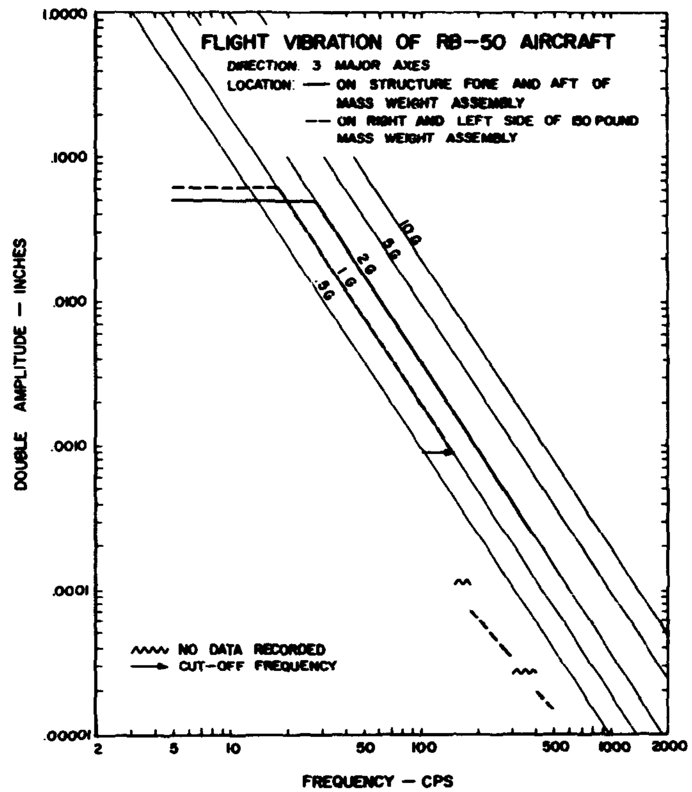

Figure 6

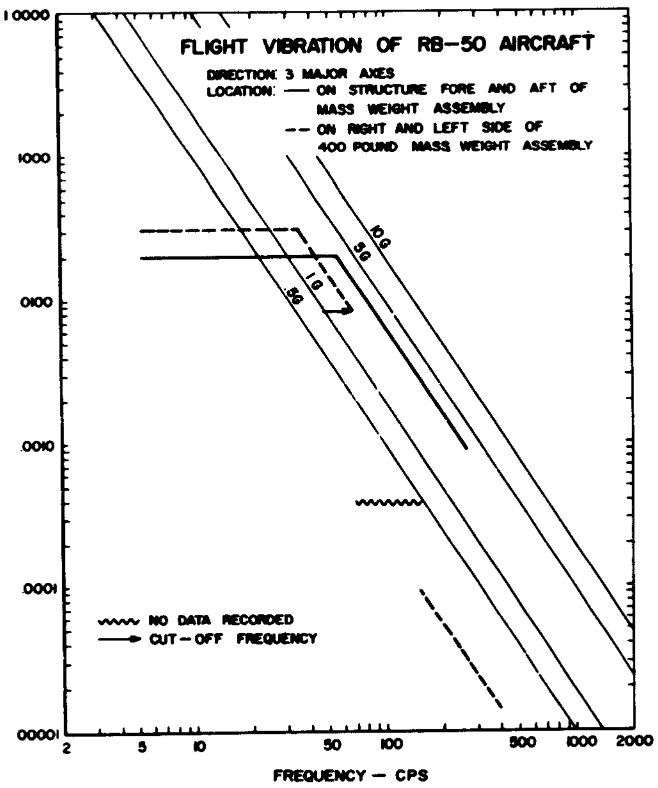

Figure 7

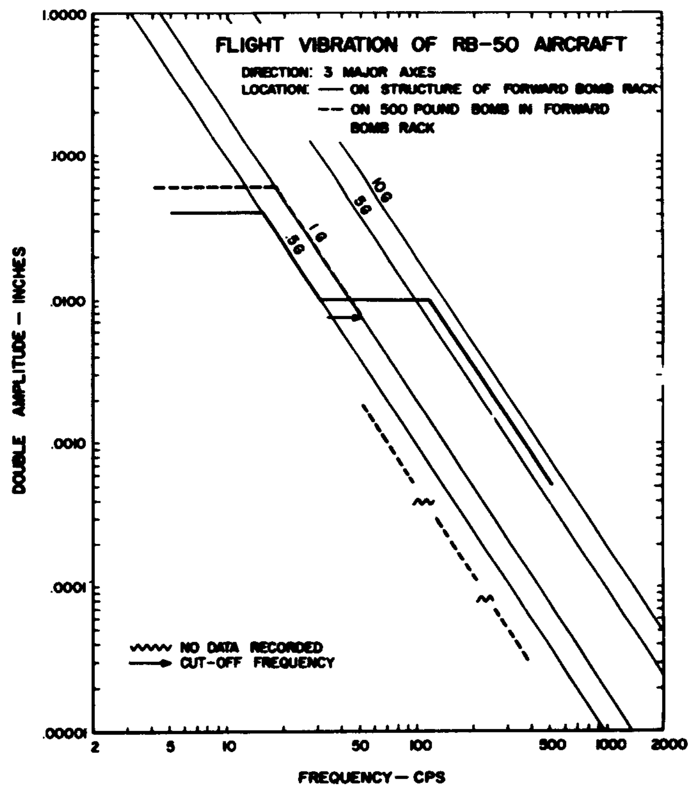

Figure 8

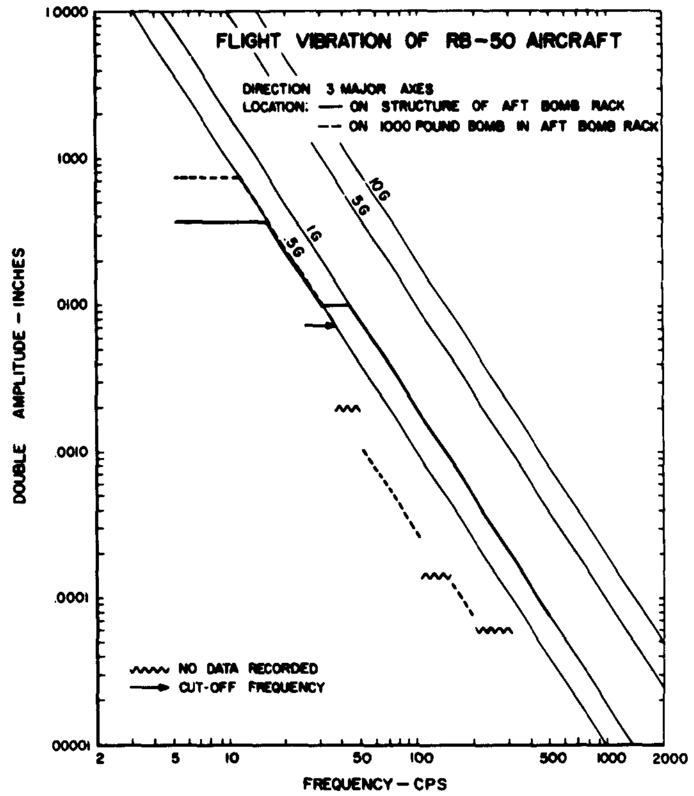

Figure 9

TABLE 2
Upper Frequency Cutoff

Test Item	Cutoff Frequency (cps)
40-pound camera in tail compartment	400
40-pound mass-weight assembly in nose section	400
100-pound camera in tail compartment	250
150-pound mass-weight assembly in nose section	200
400-pound mass-weight assembly in nose section	70
500-pound bomb in forward bomb rack	50
1000-pound bomb in aft bomb rack	35

of mounting, or the position within the aircraft, the same upper frequency cutoff would result. For example, the 40-pound camera and the 40-pound mass-weight each had a 400-cps frequency cutoff, yet they were shock-mounted and rigidly-mounted and positioned in the tail compartment and in the nose section, respectively. Hence, the change in frequency cutoff appears to be a function of weight only. In each case, the frequency cutoff of the adjacent structure exceeded that of the test item.

As the weight of the test item increases, there is an apparent trend, beginning with the plot of the 150-pound mass-weight (Fig. 6), for the item's amplitude in the low frequency regions to become increasingly larger than the amplitude of the adjacent structure. The percentage deviation of the amplitude of each of the three mass-weight assemblies from the amplitude of the adjacent structure was plotted against frequency in Fig. 11.

CONCLUSIONS AND RECOMMENDATIONS

Apparently, the upper frequency cutoff values, which produce a smooth curve when plotted against weight, are a function of weight

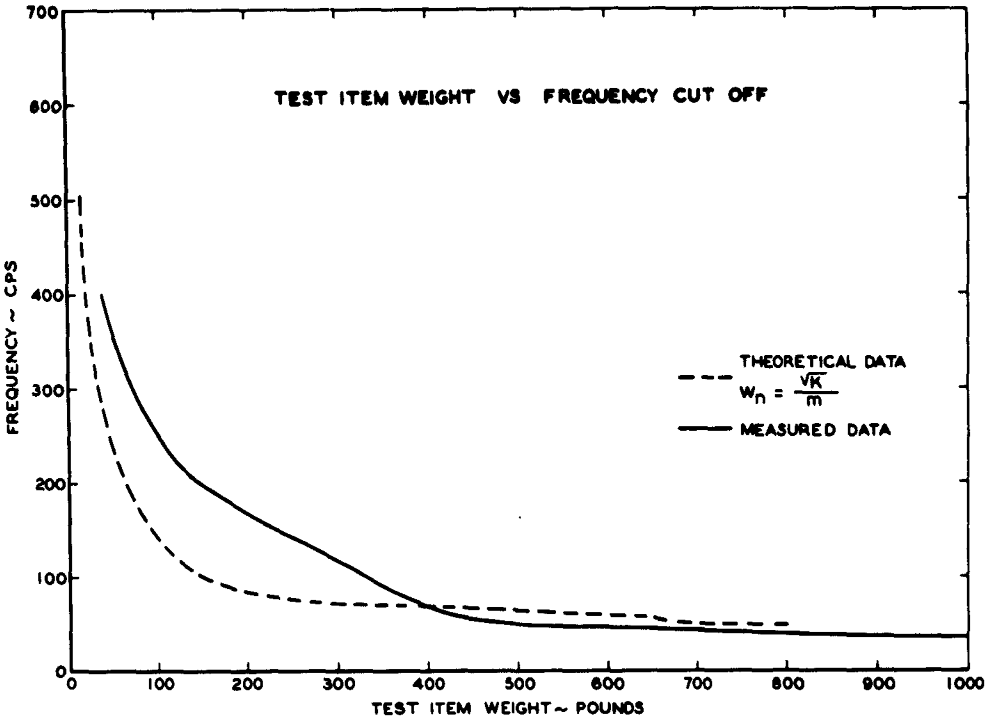

Figure 10

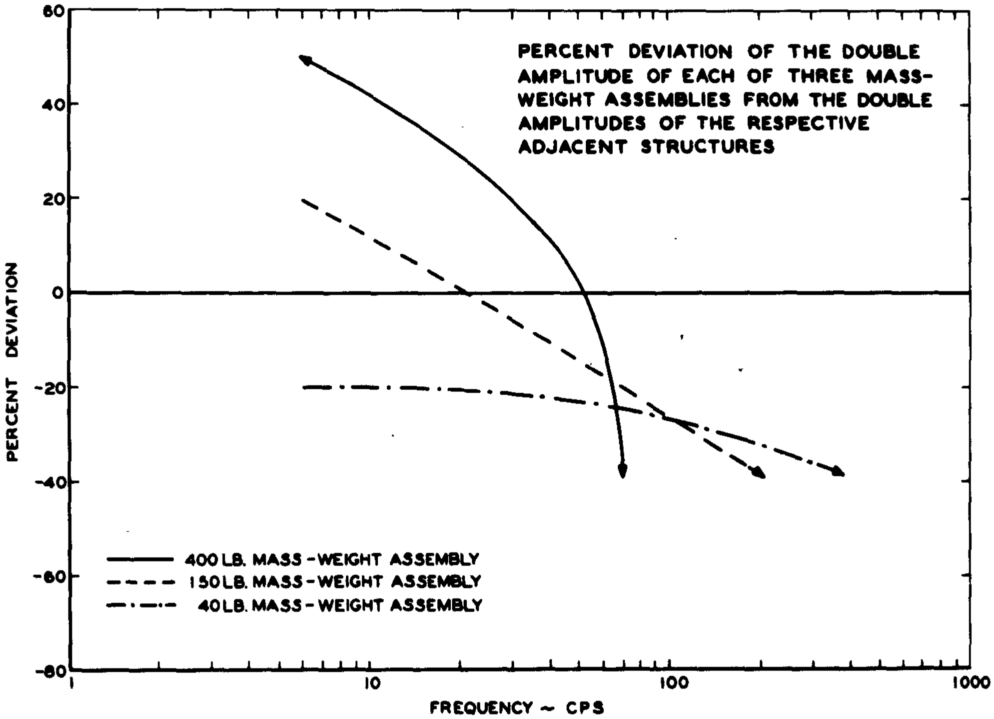

Figure 11

only. This relationship is not affected by differences in the nature of the test item, the manner of mounting the test item (shock-mounted or rigidly-mounted), or the position of the test item within the aircraft.

The amplitude of the test item seems to become progressively larger than the amplitude of the adjacent structure in the low-frequency region as the weights of the test items reach and exceed the 150-pound weight level. This phenomenon is attributed to the fact that the increased weight "tunes" the system down into the excitation region, that is, the low-frequency region of the driving function.

Although the data presented in this paper apply only to specific objects in the RB-50 aircraft, it is believed that they can be utilized more extensively to determine the effect of weight on the cutoff frequency and the low-frequency amplitude of other items in this and other types of vehicles.

General conclusions have been drawn and the curves derived from data along each major axis and under all flight conditions expected in service. Special cases were analyzed to test the trends discussed.

The application of the data in Figs. 10 and 11 to laboratory techniques would permit the conducting of effective and economical tests. It is believed that study of these dynamic characteristics can be extended to operating equipment as well as to transportation items.

These data can also be related to other problem areas in the field of dynamics, that is, to vibration propagation, vibration generation, equipment reliability, and structural loading.

* * *

SHOCK TEST METHODS VERSUS SHOCK TEST SPECIFICATIONS*

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Current shock test shortcomings are discussed and some techniques are presented to aid the designer and test engineer in evaluating component shock test data.

INTRODUCTION

Mechanical shock testing as now performed on various small components in industry is continually plagued with five major problems. These problems are inadequate test specifications, poor instrumentation techniques, inferior fixture design, generated pulse shape deviations, and improper evaluation of test data. It is because of these five problems that there is a double standard in conducting shock tests as regards the conformance of the test to the specifications. An honest effort usually is made to generate the input shock conditions within tolerances specified in the Test Requirement (TR). However, too often secondary items of concern in shock testing are totally ignored, either by accident (because of the lack of understanding of shock motion) or by intent (because of the lack of knowledge to solve the problem). These secondary items, if not considered, can have a profound effect upon the correct interpretation of the shock test results. The purpose of this paper is to point out some of the nonconformances and to discuss techniques which can be used by test and design engineers to improve the situation.

DISCUSSION OF TYPICAL SPECIFICATION SHORTCOMINGS

Consider now what might be a typical TR for a shock test. "Subject the test specimen to three half-sine shocks of 100-g maximum acceleration, 24-ms duration (measured at the 10-g acceleration level) in each direction of the three mutually perpendicular axes for a total of 18 shocks. The standard tolerances shall apply."

The information indicated is the extent of the specification. Items such as the following are often ignored in the TR, when in reality they affect proper evaluation of the test data. These items are specimen mounting requirements, accelerometer location for input shock pulse control, frequency response of the pulse monitoring equipment, incomplete functional and structural response data, "pre-shock" or "post-shock" acceleration effects, and generated pulse deviations. When items such as these are not considered, the real cause for a specimen failure in some cases might escape detection, and the input shock conditions would get the blame. These six items will now be discussed in more detail.

Specimen Mounting Requirements

If not otherwise stated, the test item should always be mounted to the shock test fixture in the same manner in which is would be mounted to the next assembly in its normal application. If the normal mounting cannot be accomplished, then a suitable substitute method must be determined. Substitute mounting techniques are probably used in the early developmental test of a new component. The substitution might consist merely of strapping the unit to the carriage instead of fastening it to a fixture by using bolts. The reason the mounting technique is important is that changes in the technique from one shock test to another can cause different responses of the component to the input shock. Refer now to Fig. 1. Considering only the small component, if the requested shock rise time is greater than five times the natural period of the thin shell subassembly (the fixture being very stiff), then

*This paper was not presented at the Sympsium.

S.C. - Small Component

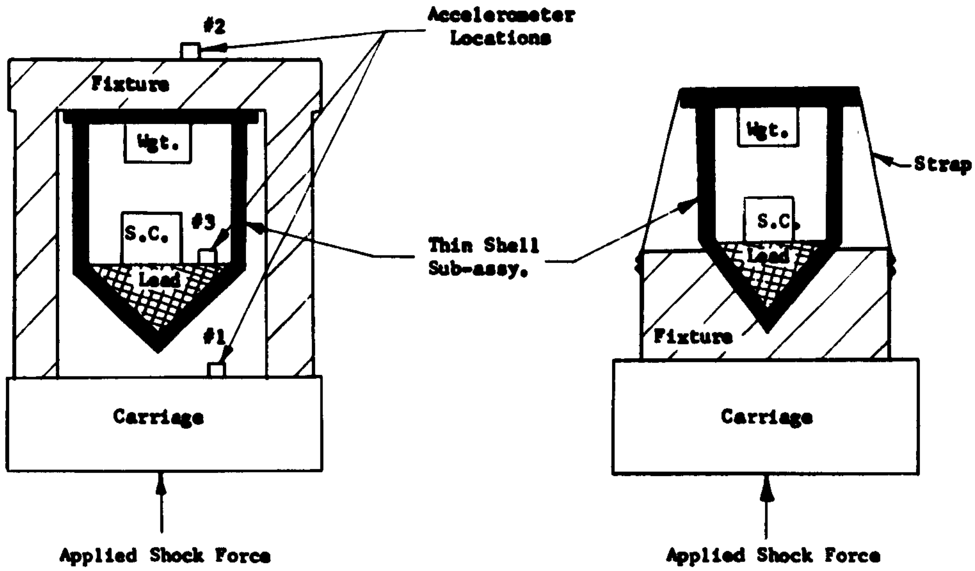


Fig. 1 - Two different mounting configurations and three accelerometer locations

either mounting configuration is adequate for good shock transmissibility. If the shock pulse rise time is less than five times the natural period of the thin shell subassembly, then the input pulse to the small component becomes significantly different from the input to the thin shell. The right hand mounting configuration would then be preferred for shock testing the small component.

Considering the entire subassembly, one can see that the point of application of a shock could be important. In the left hand configuration, the shell is in tension during the shock. In the right hand figure, the shell is in compression and possibly is buckling. The stresses in the shell undoubtedly would be different in the two cases. The shock motion of the upper weight could also be considerably different.

If the test item has long external electrical cables which are massive in size, the method by which these cables are looped or fastened to the carriage so that the shock test will not destroy the cabling must be considered. In addition, possible cable whip feeding back into the test item must be considered after the primary shock.

Input Accelerometer Location

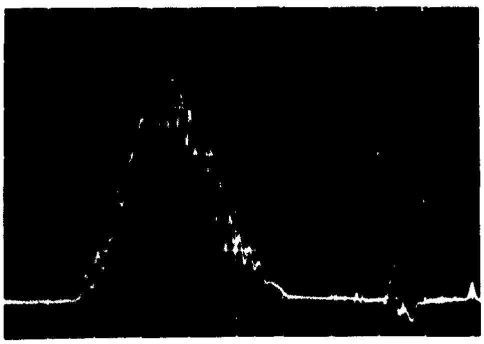
The input shock conditions to a specimen are frequently monitored by an accelerometer placed in some arbitrary location on the shock machine carriage. A shock pulse record reveals information about the shock pulse only at that particular point. The accelerometer at another spot on the carriage can record a slightly different input shock. When fixtures are added to the carriage and the accelerometer placed on the fixture, this problem of different recorded inputs can be amplified. In Fig. 1 (left-hand configuration), an accelerometer mounted on top of the fixture could easily record a different input shock from one mounted directly on the carriage. This is especially true if the subassembly is heavy and a flexible fixture is used which is eccentrically loaded. Therefore, to monitor properly an input shock, the input accelerometer should always be located close to (but not touching) the particular specimen concerned. Thus in Fig. 1, the input accelerometer for the small component should be located on the lead beside the component. If the entire subassembly is the item of concern, then the accelerometer should be located on top of the fixture.

Frequency Response of Pulse Monitoring Equipment

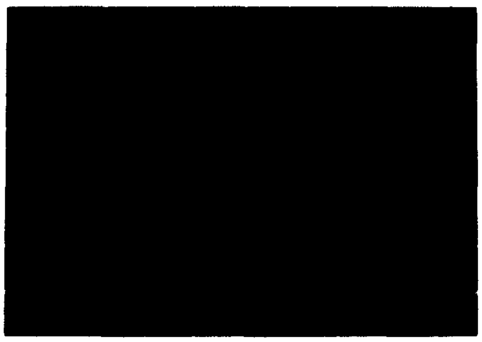
To conduct a satisfactory instrumented shock test, the engineer must know what range of frequencies should be monitored by the input accelerometer circuitry. Most shock pulses have some overriding frequencies which might be detrimental to a specimen. If the instrumentation did not record these higher frequencies, but faithfully recorded the basic shock pulse, the results of the test might be compromised.

A good example of this problem is shown in the arbitrary selection of an accelerometer to monitor a shock pulse generated by some equipment which transmits a large amount of high-frequency ringing to the test item. If the high-frequency ringing is greater than 1.5 times the accelerometer natural frequency, the signal will be greatly attenuated. Figure 2 shows a comparison of a particular shock pulse monitored by a piezoelectric accelerometer having a natural frequency of 35 kc and two strain gage accelerometers having natural frequencies of 2 kc and 850 cps, respectively. The true shock pulse obviously can be disguised, therefore the instrumentation selected should include accurate recordings of frequencies of concern other than the basic shock pulse.

(a) Piezoelectric accelerometer, natural frequency = 35 kc



(b) Strain gage accelerometer, natural frequency = 2 kc



(c) Strain gage accelerometer, natural frequency = 850 cps

Fig. 2 - Single shock pulse monitored by accelerometers having three different natural frequencies. Scope sensitivities: 25 g/major div., 2 ms/major div.

Incomplete Functional and Structural Response Data

The error that usually occurs in the functional response measurement of the test item is that the measurement is not made to the same time base reference as the input shock condition. This oversight most frequently occurs when two or more oscilloscopes are used to record the test.

In addition to a common time base reference, a structural response monitoring circuit should have the same amount of filtering as the input monitoring circuit. If these two conditions are not met, then, at best, the response measurement can only consider amplitude, ignoring completely any phase shift from the input. Phase differences are lost completely without a common time reference. If the response monitoring circuit has more filtering than the input circuit, the recorded phase shift of the response from the input will have a greater lag than the actual phase shift that exists.

"Pre-Shock" and "Post-Shock" Acceleration Effects

The input shock pulse specified in the TR is not the only input acceleration experienced by the specimen from the start to the conclusion of a shock test. There will also be secondary accelerations. Since the carriage on which the test item is mounted starts from rest and ends at rest (if the test item is to be recovered), accelerations have to occur in more than one direction. At Sandia Corporation, secondary acceleration levels during a shock test generally are less than 20 percent of the requested peak acceleration and so are often ignored. Even with this restriction, the error in maximum response acceleration of the test item might be greater than 25 percent. If secondary accelerations are ignored, therefore, the test item might still be responding from a preliminary acceleration at the time the primary shock is applied or else might respond adversely to a secondary shock pulse after the primary shock is over. From this standpoint, the response of a test item to the generated shock should be considered.

If the test item can be considered a single-degree-of-freedom, linear, undamped system, there exists a graphical technique for one to determine the acceleration-time response of the item to any kind of transient input. This graphical technique uses the so-called "phase-plane" method. The graphical method is especially suited for analyzing generated input shock pulses that cannot easily be described mathematically. Points taken directly from one phase-plane plot will locate primary and residual responsive amplitudes with respect to time for any one specimen involved in the test. This method will work for any transient disturbance where the amplitude is acceleration, velocity, or displacement. The theory behind the method can be found in Refs. 1 and 2 and will not be discussed here. The technique requires only the use of graph paper, protractor, compass, straight-edge, and knowledge of the specimen's natural period.

Figure 3 shows how a phase-plot of a single-degree, linear, undamped system is generated from a step-wise transient input acceleration. The input is shown as a dotted line. The centers of each circular segment of the phase-plane are determined by the acceleration level of each step of the transient input, namely, a_0, a_1, a_2, a_3, and a_4. The radius, r_1, of the first circular segment is determined by the initial starting conditions. (In the figure it is assumed that the system initial acceleration and initial jerk are zero; however, the same procedure would be followed for any other starting conditions.) This radius, r_1, is a rotating vector for the time interval, t_1, of the first step-wise input a_1. The counterclockwise angle through which the rotating radius vector turns is dependent upon the natural frequency, p, of the system and the time interval of the step. The conditions of motion of the system at the end of the first-step interval, 1, become the new starting conditions of the system for the second-step interval of the input. A new center, a_2, and a new radius, r_2, are now used. The second radius vector rotates through the second angle which is proportional to the second time interval, t_2. The end conditions of the second interval, 2, become the starting conditions for the third-step interval, and so on, and the procedure is repeated as often as is necessary. One can easily see, therefore, that this procedure can be applied to any transient input shape as long as the shape can be approximated by steps and each step has the same area as that portion of the input transient.

After the phase-plane plot is completed, points representing various responsive accelerations from the phase-plane plot can be projected back to the acceleration-time history plot which is superimposed on the transient input.

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[1]Harris, C. M., and Crede, C. E., Shock and Vibration Handbook (McGraw-Hill Book Company, Inc., New York, 1961), Vols. I and II.

[2]Timoshenko, S., Vibration Problems in Engineering (D. Van Nostrand Company, Inc., 1955).

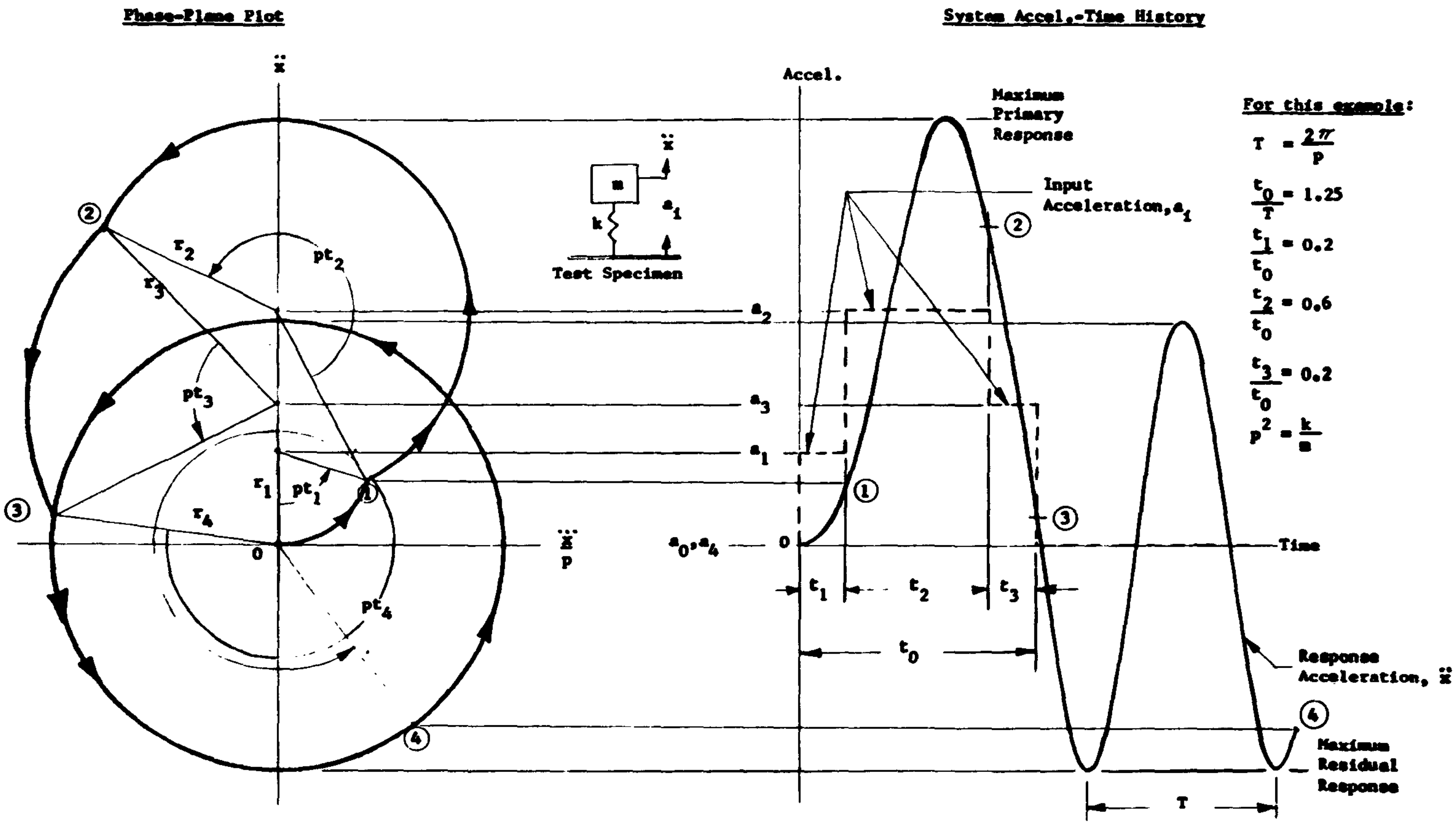

Fig. 3 - Response of single-degree, linear, undamped system to transient input pulse

Thus, the entire acceleration-time history of the responding spring mass system is shown in the right-hand side of Fig. 3.

Figure 4 is a phase-plane plot and a normalized plot of the acceleration-time history of a single-degree, linear, undamped system, having a natural frequency of 1 kc, subjected to a 1000-g, 1-ms, symmetrical triangle pulse. The assumption is made that this pulse is generated on a free-fall machine which has no brakes. Rebound of the machine permits a second impact to occur. The first impact is seen to cause residual vibration of the specimen. While the specimen is still vibrating, the second impact occurs after a certain interval of time. The resulting residual vibration is greatly increased in this instance. (If a different time interval was involved for the same specimen, the amplitude of vibration would have been different.) By converting Fig. 4 to the conditions of the example, it is evident from the first impact that the primary maximum response of the system would be +1500 g (point 1), while the maximum residual response would be ±1250 g (points 3 and 5). The time interval between impacts is some multiple integer of 1 ms. From the second impact, the maximum primary response of the system is +2700 g (point 15), while the residual response is ±2500 g (points 17 and 19). If the specimen had failed from the second impact and only the first impact had been instrumented, improper evaluation of the test would have resulted.

Generated Pulse Deviation

Most shock producing equipment will never generate a shock pulse which is identical to a requested shock pulse as it is now specified. For the most part, the specified peak acceleration and duration can be satisfied, but the generated shape does not approximate very closely the requested shape. Even though the pulse shape in shock testing can have a very important effect upon the response of the specimen to the shock, this nonconformance or deviation from the specification is usually accepted by the designer without question. The author feels that if the generated pulse is too different from the requested pulse, a check should be made in regard to the response of the specimen from the generated pulse and the information compared to the specimen response for the requested pulse. The check can be made by using the "phase-plane" method mentioned previously if a single-degree, linear, undamped system can be assumed. (Refer again to the phase-plane plots shown in Figs. 3 and 4. The intersections of the phase path with the $\ddot{x}$-axis determines the maximum primary and residual acceleration of the system to any generated transient. The system acceleration-time history does not have to be known.) The comparison will indicate whether the generated pulse shape results in either an undertest or overtest for the particular specimen. The test parameters could then be changed accordingly. The phase-plane method is a very quick way to obtain an immediate comparison of the maximum response of two or three systems to shock spectra for standard shapes when computers are not readily available.

A method is available at Sandia Corporation to compare records of the generated pulse to requested pulse shapes. The generated pulse can be catalogued to best approximate one of several pulse shapes standardized by Sandia Corporation by means of a nomograph (Fig. 5)* designed by the author. (The nomograph also has other uses which will not be discussed here.) The specimen response to the catalogued pulse is then compared with the specimen response to the requested pulse.

Perhaps, an example is the best way to see how the nomograph is used to catalogue a pulse shape. Assume that a TR specifies a half-sine shock pulse be generated of 100-g peak acceleration, and 24-ms duration (measured at the 10-g acceleration level). A tolerance of ±15 percent on all recorded pulse parameters is permitted.

The acceleration-time history in Fig. 6 is obtained from a shock machine which has restrictions in regards to shock pulse shaping. To comply with the TR, the magnitudes of the variables shown in Fig. 6 must be compared to the requested values (Table 1).

TABLE 1
Recorded Pulse Comparison to Half-Sine Pulse Shape

Pulse Parameter	Requested Half-Sine Values (Nomograph)	Recorded Values	Percent Deviation
A_p (g)	100	100	0
t_{10} (ms)	24	24	0
t_0 (ms)	25	36	+44
t_r (ms)	8.2	10	+22
ΔV (fps)	52	40	-23

*Copies, 11 x 14 in size, are available upon written request to the author.

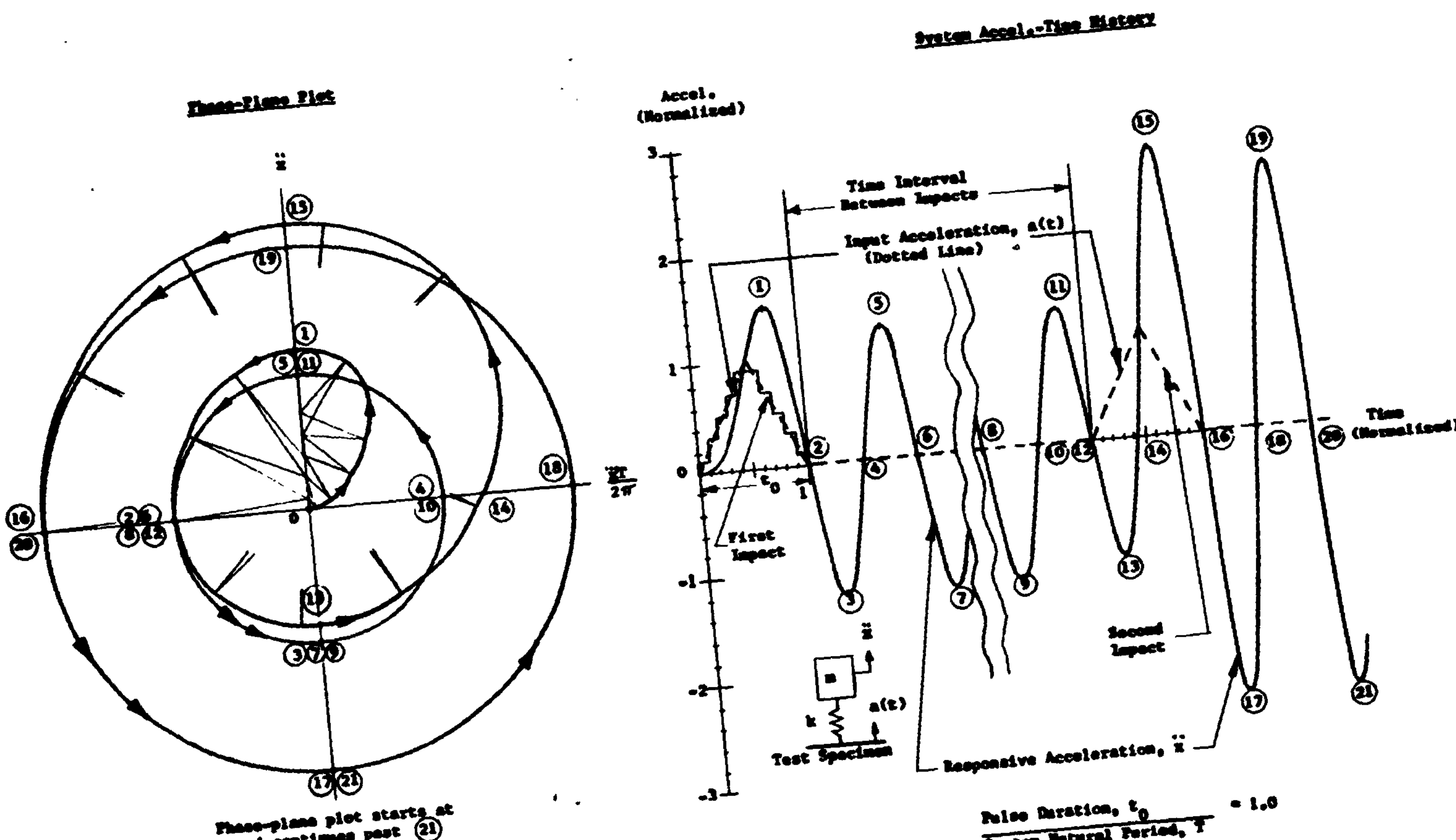

Fig. 4 - Response of single-degree, linear, undamped system to an impact shock test where second impact can occur

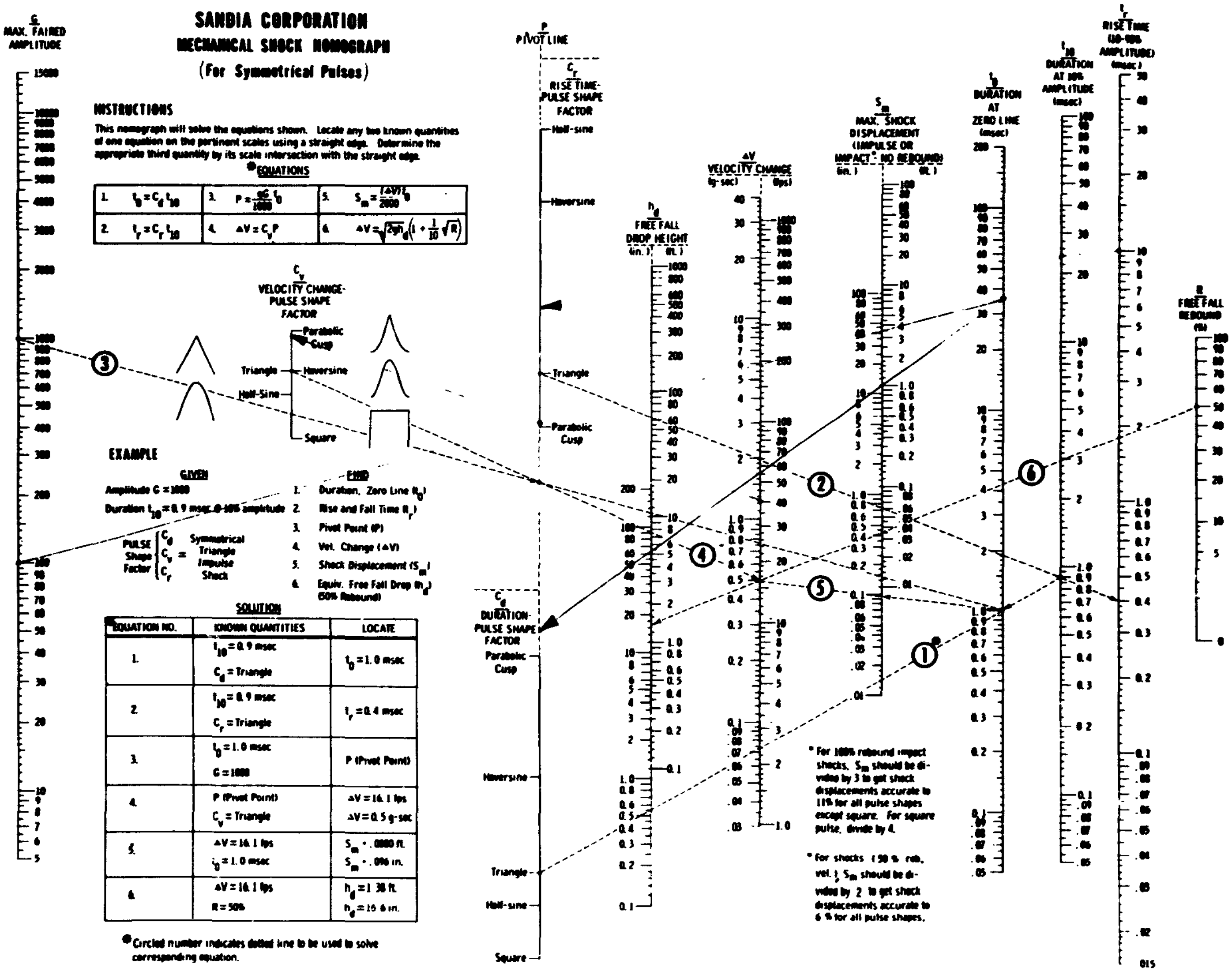

Fig. 5 - Mechanical shock nomograph

Scope Sensitivities
20g/div.
4ms/div.

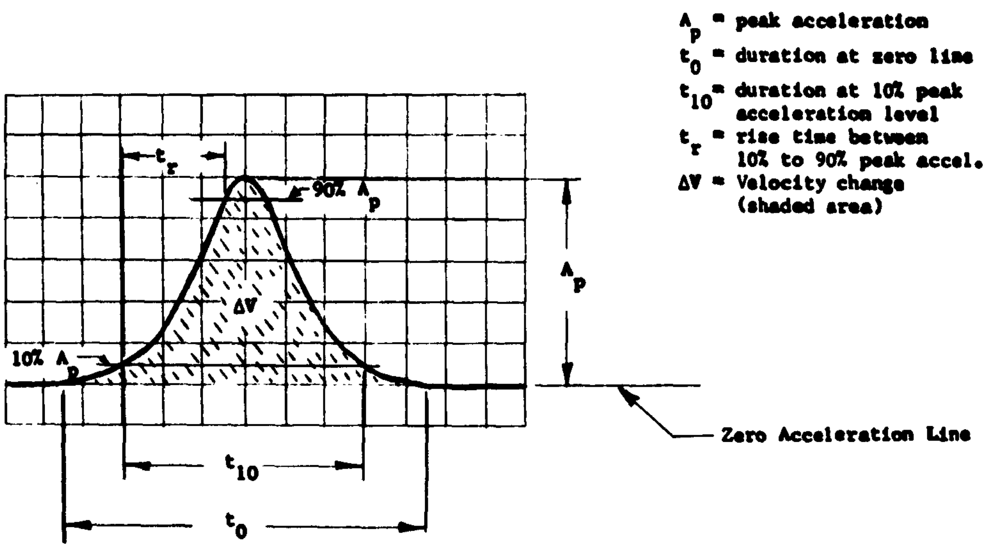

Fig. 6 - Recorded shock pulse

From Table 1 it is obvious that the equipment available for generating the requested pulse cannot do so in this example. If a different pulse shape had been requested, then perhaps the tolerances could have been met. The generated pulse will now be catalogued to determine a better shape in order to meet the tolerances. Cataloguing consists of inserting values from the recorded picture (listed in Table 1) into the appropriate equations of the nomograph, Fig. 5. For this example, the following checks are made:

- Comparison of the two durations, t_0 and t_{10}. (Use of Eq. 1 in nomograph indicates this recorded pulse shape best approximates a parabolic cusp at the lower acceleration levels.)

- Comparison of rise time, t_r, and duration, t_{10}. (Use of Eq. 2 in nomograph indicates the recorded pulse shape best approximates a triangle, the haversine and parabolic cusp shapes being possible considerations.)

- Examination of the recorded velocity change, ΔV, peak acceleration, A_p, and duration, t_0. (Using Eqs. 3 and 4 in the nomograph, the recorded pulse shape best approximates a parabolic cusp.)

Table 2 compares the recorded pulse shape to a 100-g parabolic cusp, 24-ms duration (at 10-g acceleration level). From this analysis the shape of a parabolic cusp instead of a half-sine pulse would have to be requested before the shock pulse generated by this machine could satisfactorily meet the tolerances in the TR.

However, to insure an equivalent shock test on this machine, a check must be made of the specimen response to the catalogued pulse (parabolic cusp), and this response must be compared to the requested half-sine pulse response. Assume that the specimen in a single-degree, linear, undamped system having a natural period, T, of 25 ms (40 cps). The requested test in the TR was designed so that a near maximum response of the specimen would occur. In Fig. 7 shock spectra are shown for half-sine shock pulses. (Ref. 3.) In this example the ratio of requested pulse duration (zero line) to specimen natural period is 1. The primary response of the specimen should then be 170 g for 100-g input (residual response ±135 g).

[3]Jacobsen, L. S., and Ayre, R. S., Engineering Vibrations (McGraw-Hill Book Company, Inc., New York, 1958).

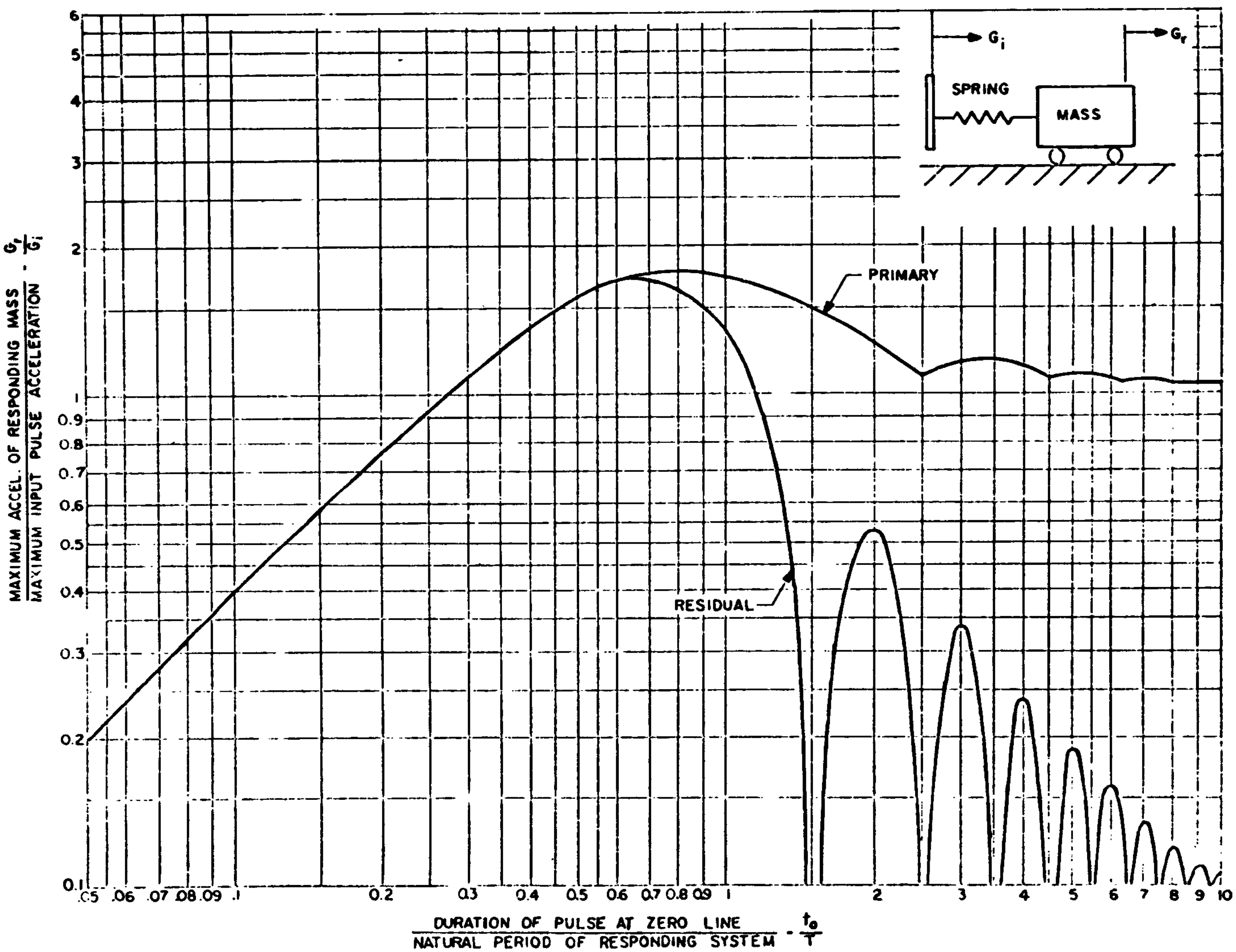

Fig. 7 - Primary and residual shock spectra for undamped single-degree linear system subjected to half-sine pulse

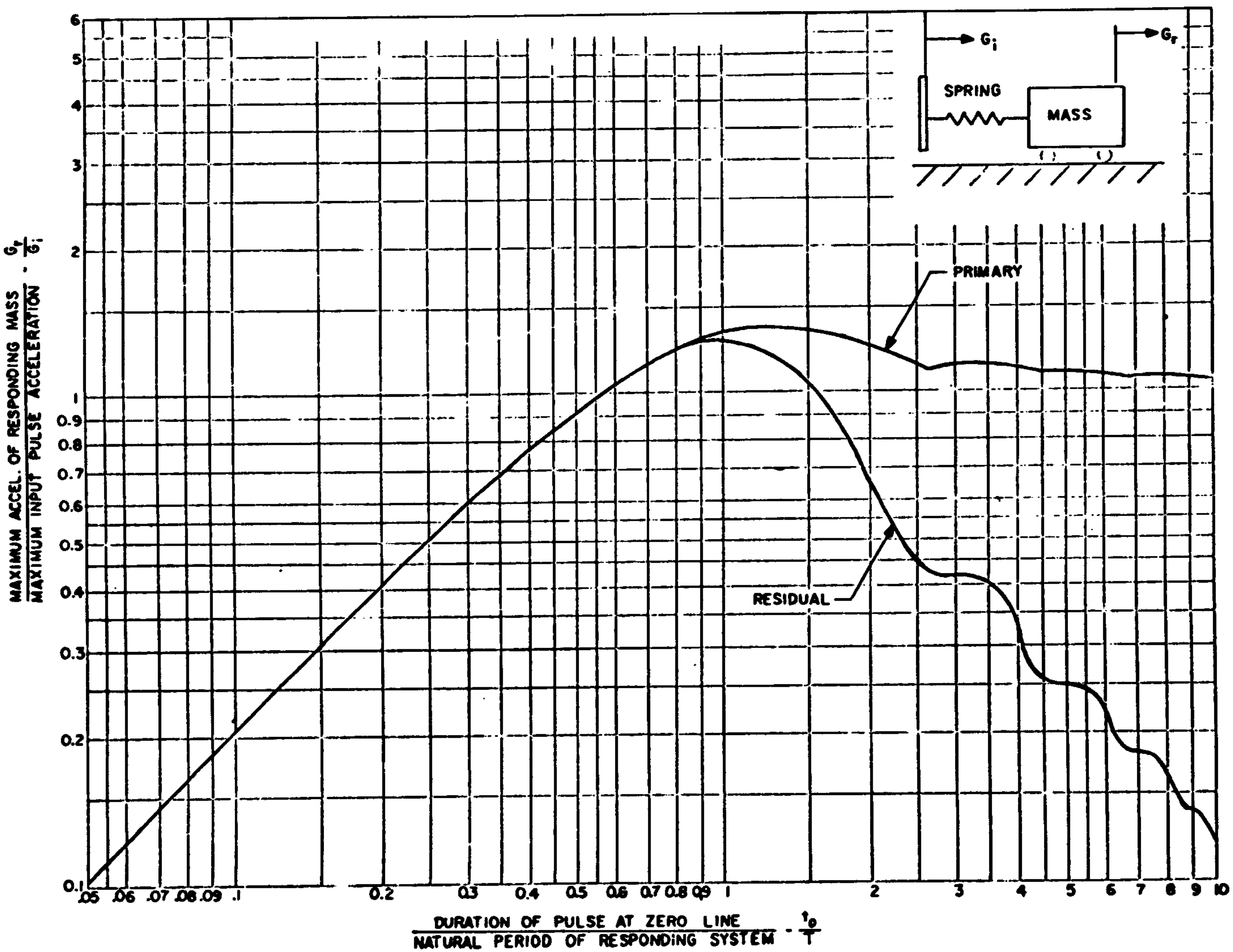

Fig. 8 - Primary and residual shock spectra for undamped single-degree linear system subjected to parabolic cusp pulse

TABLE 2

Recorded Pulse Comparison to Parabolic Cusp Pulse Shape

Pulse Parameter	Recorded Pulse Values	Parabolic Cusp Specification (Nomograph)[a]	Percent Deviation
A_p (g)	100	100	0
t_{10} (ms)	24	24	0
t_0 (ms)	36	34	+6
t_r (ms)	10	11.2	-11
ΔV (fps)	40	37	+8

[a]Verification of these numbers is left to the reader.

The specimen response to the catalogued parabolic cusp, 100 g, 36 ms at zero line, can be found from Fig. 8. (The ratio $t_0/T = 1.44$.) The primary response of the specimen to the catalogued pulse is 135 g (residual response ±110 g). From this analysis the generated pulse will cause the specimen to be undertested with respect to the original request.

Therefore, if this machine is to produce a specimen response similar to that which occurred for the original, half-sine pulse TR, a new TR should be written requesting (for the particular machine involved) a parabolic cusp, 126 g, 24-ms duration at the 13-g acceleration level. The specimen maximum acceleration response would then be 170 g (primary), ±137 g (residual).

CONCLUSIONS

This paper has attempted to show how to improve shock test specifications and how test procedures can compromise the test results. Techniques to aid the designer and test engineer in evaluating component shock test data were presented.

* * *

IMPEDANCE CONSIDERATIONS IN VIBRATION TESTING*

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The assumption of negligible specimen impedance inherent in the motion input approach to vibration testing is often unsatisfied. In this paper the significance of this assumption is illustrated by considering the relationship between the operational environment and a motion input test environment for a spacecraft-launch vehicle system in which the spacecraft impedance is appreciable.

INTRODUCTION

The most widely used procedure for testing the ability of a structure or equipment to withstand a vibration environment is to subject its base or attachment points to a prescribed oscillating motion input, at test levels derived from an envelope of measured vibration data. Inherent in this testing approach is an assumption that the impedance of the test specimen is always much smaller than that of its supporting structure; or equivalently, that the measured vibration levels on which the test specification is based will not be changed appreciably by the reactions of the test specimen when it is finally installed on its supporting structure.

This motion input approach to vibration testing has been criticized in the past[1,2] on the grounds that the assumption of negligible specimen impedance is often unsatisfied when dealing with large or even moderately large structures, and the resulting tests are generally overconservative by large factors. As yet, however, a widespread appreciation of this concept is not evident, and motion input test requirements continue to appear even for very large specimens. In this paper, the significance of the small impedance assumption is illustrated by considering the relationship between the operational environment and a motion input test environment for a spacecraft-launch vehicle system in which the spacecraft impedance is appreciable.

MECHANICAL IMPEDANCE

The mechanical impedance, z_{ij}, of a structure is conventionally defined as the complex ratio of the harmonic exciting force to the resulting harmonic velocity,[3] thus

$$z_{ij} = \frac{F_j}{\dot{q}_i}, \tag{1}$$

where the subscripts i, j indicate particular coordinates on the structure. If the velocity is measured at the point of excitation (i.e., $i = j$), the resulting impedance is known as a direct or driving point impedance; otherwise, the designation transfer impedance is used.

Mechanical impedance is a convenient measure of the resistance of a structure to vibration, the impedance being high for a structure that is inherently difficult to excite, and low for a structure that is readily excited. For typical lightly damped structures the impedance varies

*This paper was not presented at the Symposium.
[1]Blake, R. E., "The Need to Control the Output Impedances of Shock and Vibration Machines," Shock and Vibration Bulletin No. 23 (June 1956).
[2]Blake, R. E., and Belsheim, R. O., "The Significance of Impedance in Shock and Vibration," ASME Colloquium on Mechanical Impedance Methods, ASME (1958).
[3]Crandall, S. H., "Impedance and Mobility Analysis of Lumped Parameter Systems," ASME Colloquium on Mechanical Impedance Methods, ASME (1958).

sharply as a function of frequency over a range of about two orders of magnitude. Minimum or zero values of impedance correspond to resonant frequencies, whereas maximum or infinite values of direct impedance correspond to the antiresonant frequencies associated with the driving point under consideration.

Impedance methods are especially useful in analyzing the coupling of two systems having known characteristics. For systems coupled at a single coordinate it can readily be shown[3] that the direct impedance at the interface of the coupled system, z_{TOT}, is simply the sum of the direct impedances of the two subsystems, that is,

$$z_{TOT} = z_1 + z_2 . \tag{2}$$

For undamped systems, the resonant frequencies occur when the impedance vanishes, therefore

$$z_{TOT} = z_1 + z_2 = 0 , \tag{3}$$

or

$$z_1 = -z_2$$

becomes the frequency equation for the coupled system.

The relative impedance requirement that must be satisfied for a motion input specification to be a realistic test is evident from these simple expressions. Assuming that the response measured at the interface of the larger of the two systems (indicated by subscript 1) is essentially the same with or without the smaller system, requires that z_{TOT} be approximately equal to z_1, or equivalently, that z_1 be much greater than z_2. If z_{TOT} is considerably different from z_1, both the resonant frequencies and the interface vibration levels will be affected.

IMPEDANCE EFFECTS FOR A TYPICAL SPACECRAFT-LAUNCH VEHICLE SYSTEM

The system used for illustration in this section consists of the Orbiting Astronomical Observatory, a 3300-pound satellite, being developed by the Grumman Aircraft Engineering Corporation for the Goddard Space Flight Center of NASA, and its Atlas-Agena B launch vehicle. As a part of the spacecraft development program, the direct impedances of both the spacecraft and the launch vehicle were calculated in the axial direction at the interface. These calculations were based on lumped parameter representations and assumed structural damping factors of 0.04. Curves of the resulting impedances are presented in Fig. 1 for frequencies up to 500 cps. Although the upper portions of this frequency band are subject to considerable inaccuracy because of the limitations of the mathematical representation, the system is only being regarded as typical, and the general discussion will not be affected by these inaccuracies.

The curves of Fig. 1 reveal that the launch vehicle impedance is much higher than the spacecraft impedance over most of the frequency range; however, there are also several ranges in which the spacecraft impedance is as high as or higher than that of the launch vehicle. Since flight vibration is primarily a resonant phenomenon, measurements made on the launch vehicle without a spacecraft or with a small spacecraft would indicate maximum vibration at the launch vehicle resonances, or frequencies of minimum impedance, and much lower levels at other frequencies. Typical motion input vibration specifications, which are based on envelopes of peak vibration levels, are therefore representative of minimum values of launch vehicle impedance; significantly, it is at these minimum values that Fig. 1 shows the spacecraft impedance to be comparable to or greater than that of the launch vehicle. The fact that the launch vehicle impedance is much larger in other ranges is of little importance since no appreciable vibration occurs in those ranges.

During a motion input vibration test, the largest amplifications of the input motion, and thus the most severe conditions, occur at the antiresonant frequencies of the spacecraft, or frequencies of maximum spacecraft impedance. This is shown in Fig. 2, for the system under consideration, where the ratio of the response at the top of the spacecraft to the input motion is plotted as a function of frequency. A maximum amplification of 27 occurs at 53 cps, the first spacecraft antiresonant frequency.

By requiring that the input motion at the spacecraft antiresonant frequencies be representative of the launch vehicle resonant response, the motion input test anticipates a condition in which a launch vehicle resonance coincides with a spacecraft antiresonance. It follows, then, that in those frequency ranges for which the motion input test is most severe, the resulting spacecraft response is realistic only if the maximum spacecraft impedance is much less than the minimum values of impedance associated with neighboring launch vehicle resonances. If reference is again made to Fig. 1, it

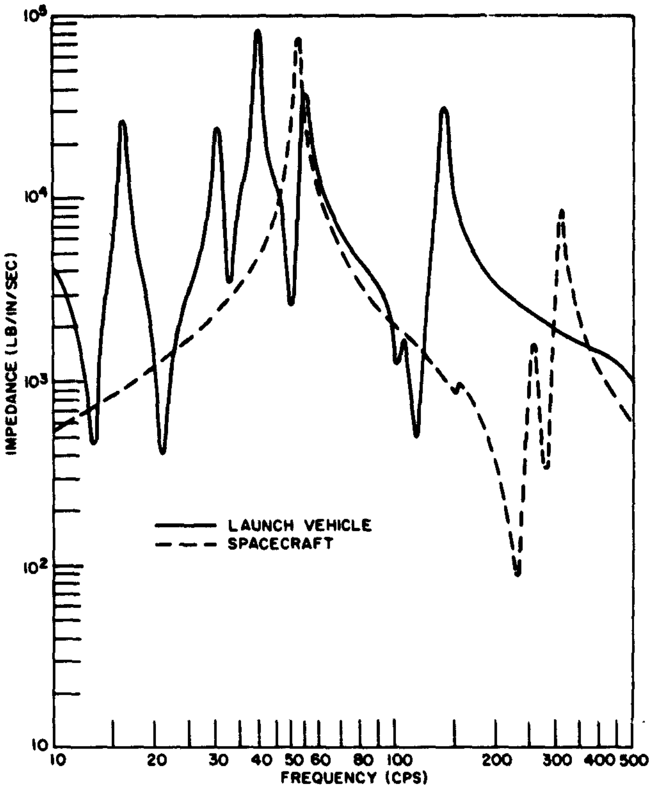

Fig. 1 - Direct impedance curves for typical spacecraft and launch vehicle

is seen that the impedance of the first spacecraft antiresonance (53 cps) is thirty times as large as the impedance of the nearest launch vehicle resonance (50 cps). Furthermore, the impedance at every one of the spacecraft antiresonances exceeds that of any neighboring launch vehicle resonance. It must therefore be concluded that for this configuration the spacecraft impedance cannot be considered small relative to that of the launch vehicle. The reaction forces generated by the vibrating spacecraft may be expected to have an appreciable effect on the vibration levels at the interface. The large margin by which the small impedance assumption is invalid in this case casts serious doubts as to its validity in many other cases, and provides an effective demonstration of the fact that it may be seriously erroneous to assume that a test specimen with relatively small mass also has relatively small impedance.

The conditions under which a motion input test is strictly valid have been discussed; next, the results of misapplying such a test by imposing it on systems having appreciable impedance must be considered. First, the changes in resonant frequencies due to the coupling of two systems will be discussed.

For systems linked at a single coordinate, a graphical solution of Eq. 3 provides a convenient method of obtaining the coupled system resonant frequencies. If the undamped impedance of the first system and the negative of the undamped impedance of the second system are plotted on the same graph, the intersections of the two impedance curves will occur at the resonant frequencies of the combined system. This plot is presented for our typical spacecraft-launch vehicle system in Fig. 3. It is seen that the addition of the spacecraft has resulted in nearly doubling the number of resonances in the frequency range below 500 cps, with the original resonances of the launch vehicle (which occur where the launch vehicle impedance curve crosses zero) being shifted in frequency to varying extents. Since flight vibration is primarily resonant, it will be at these new resonant frequencies, rather than the

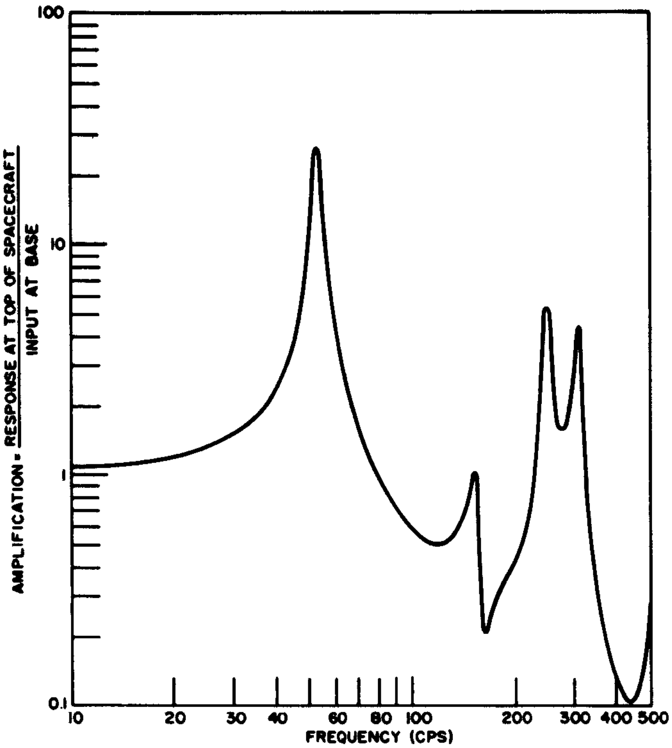

Fig. 2 - Response at top of spacecraft for a motion input at the base

original launch vehicle resonances, that the most severe vibration will occur.

Since the form shown by the direct impedance curves of Fig. 3 is characteristic of all undamped systems, some comments concerning their general behavior appear appropriate. In all mechanical systems the direct impedance alternates between resonances and antiresonances, with a change of sign occurring at each of these frequencies. When two mechanical systems are linked at a single coordinate, an antiresonance of the combined system occurs for the coupled coordinate at each antiresonance of either of the two subsystems, and again a resonance occurs between every two antiresonances. If one of the two systems has a generally small impedance compared to that of the other, the resonances of the combined system will fall near the resonances of the system with the larger impedance and near the antiresonances of the system with the smaller impedance. The validity of these comments can be deduced from the characteristic behavior of these impedance curves. For further information the reader is referred to the comprehensive discussions of Refs. 4 and 5.

The change in resonant frequencies due to the installation of a spacecraft having significant impedance has now been described; the next factor to be considered is the effect of the spacecraft reaction forces on the vibration levels at the interface. A detailed description of the actual interface flight environment for the typical launch vehicle being considered is not available for this discussion, and the actual excitation causing flight vibration is extremely complex and not known in sufficient detail to permit a calculation of these levels. The discussion will therefore be based on the vibratory response of the launch vehicle to a simple harmonic excitation. Since direct impedance curves are

[4]Bishop, R. E., and Johnson, D. C., The Mechanics of Vibration (Cambridge University Press, 1960).

[5]MacNeal, R. H., "Vibrations of Composite Systems," California Inst. of Technology Report No. 4 for ARDC (1954). (Astia AD No. 63118.)

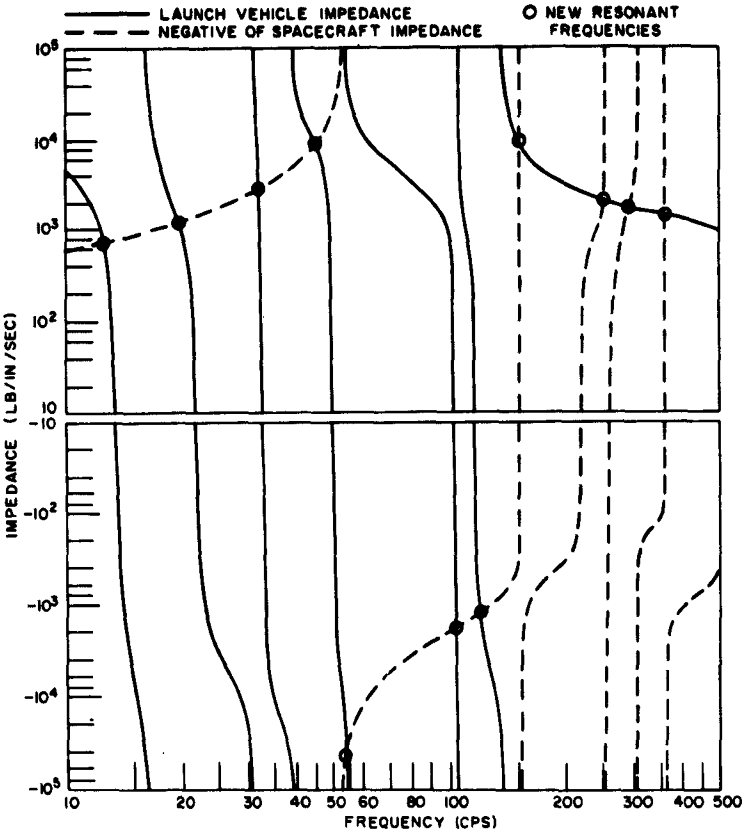

Fig. 3 - Undamped direct impedance curves for typical spacecraft and launch vehicle showing new resonant frequencies of combined system

available for the interface, this becomes the most convenient location to apply the excitation.

The frequency range of greatest interest for this discussion lies in the vicinity of the first spacecraft antiresonance, where an amplification of 27 occurs during a motion input test (see Fig. 2). This is the only spacecraft antiresonance in the lower frequency range at which significant amplification occurs.

The response curves for this discussion are most conveniently presented in terms of mobility, which is the inverse of mechanical impedance. Mobility, or the response to a unit exciting force, is plotted in curve 1 of Fig. 4 for the typical launch vehicle in the range around 53 cps, the first spacecraft antiresonant frequency. The 50-cps launch vehicle resonance is seen as a peak in the mobility curve.

If this response is now regarded as measured vibration data, and the conventional approach to establishing a conservative vibration test is followed, a straight line would be drawn as shown by curve 2 of Fig. 4, through or above the peak vibration levels observed. This would then become the test input motion for the spacecraft.

The response at the top of the spacecraft to this input motion is plotted in the third curve of Fig. 4, and it is seen that the spacecraft vibration environment becomes 27 times as severe as the original launch vehicle environment. It is not at all uncommon to proceed from this point by assuming that the spacecraft represents a large impedance to all subsystems mounted within it, and to provide an envelope of the levels measured during the vibration test as an input to these subsystems. In this manner, it is possible to "pyramid" the overall amplifications to extremely high and generally unrealistic values.

Next, it is necessary to determine the actual spacecraft response when the coupled

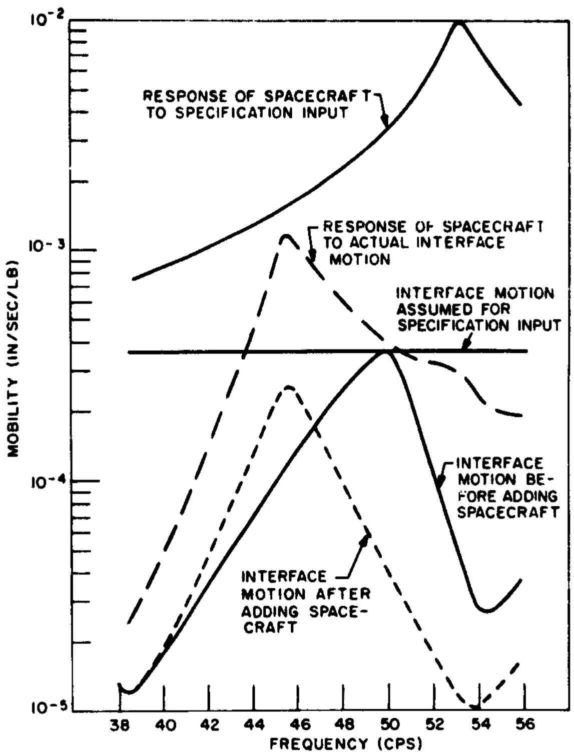

Fig. 4 - Spacecraft and launch vehicle response (mobility) curves

spacecraft-launch vehicle system is acted on by the same excitation that caused the interface motions shown by curve 1, Fig. 4. Prior to the attachment of the spacecraft, the launch vehicle impedance at the interface is

$$z_1 = \frac{F}{\dot{q}} . \tag{4}$$

After the spacecraft is attached, the interface impedance becomes the sum of the spacecraft and launch vehicle impedances (from Eq. (2)), that is,

$$z_{TOT} = z_1 + z_2 = \frac{F}{\dot{q}'} , \tag{5}$$

where $\dot{q}'$ is the new interface motion. Dividing Eq. (4) by Eq. (5) and rearranging, yields the new interface response in terms of the original interface response and the impedances of the spacecraft and launch vehicle,

$$\dot{q}' = \frac{z_1}{z_1 + z_2} \dot{q} . \tag{6}$$

If this calculation is carried out, with due regard for the phase angles of the impedances, the interface response shown in curve 4, Fig. 4 results. It is seen that at the 53 cps spacecraft antiresonance the interface vibration level has been reduced appreciably from an already low values, and that a new resonant peak has appeared at 45.5 cps.

The response at the top of the spacecraft corresponding to this new interface motion is shown in curve 5 of Fig. 4. The maximum spacecraft vibration levels are seen to be significantly lower on the actual coupled system than they are during the vibration test. For the coupled system the most severe spacecraft vibration levels are only 3 times the highest original launch vehicle levels, as compared to 27 times these levels during the vibration test. The motion input test in this case is therefore conservative by a factor of nine.

A widely held belief which has been demonstrated to be valid for a simple system[6] is that the spacecraft will experience the most severe vibration if a coincidence occurs between a spacecraft antiresonance and a launch vehicle resonance. In order to consider the result of this eventuality, the previous calculations were repeated with the spacecraft impedance curve shifted slightly to the left to bring about a coincidence between the 53-cps spacecraft antiresonance and the 50-cps launch vehicle resonance. The set of curves corresponding to this calculation are shown in Fig. 5. The interface vibration levels for the coupled system now show a pronounced dip at what was previously the frequency of maximum vibration, the new response being only 1/25 of the original. On both sides of this dip are the new resonant frequencies of the coupled system. The response at the top of the spacecraft is again much less on the coupled system than it is during the vibration test, and a comparison of the spacecraft vibration levels in Figs. 4 and 5 discloses that there is little difference between the two. Thus, even in the "worst case" the motion input test remains conservative by a factor of nine for this spacecraft-launch vehicle combination.

[6]Blake, R. E., and Ringstrom, T., "The Influence of Mass and Damping on the Response of Equipment to Shock and Vibration," Shock and Vibration Bulletin No. 28, Part IV (August 1960).

A MODIFIED TEST PROCEDURE

It has been shown in the previous sections that the impedance of a test specimen must be small compared to that of its supporting structure, if a motion input requirement is to be a valid test. If the impedance of a specimen is significant, the reaction forces generated by its vibration can have an appreciable effect on both the resonant frequencies and the vibration levels of its supporting structure. Under these conditions it is no longer realistic to consider an

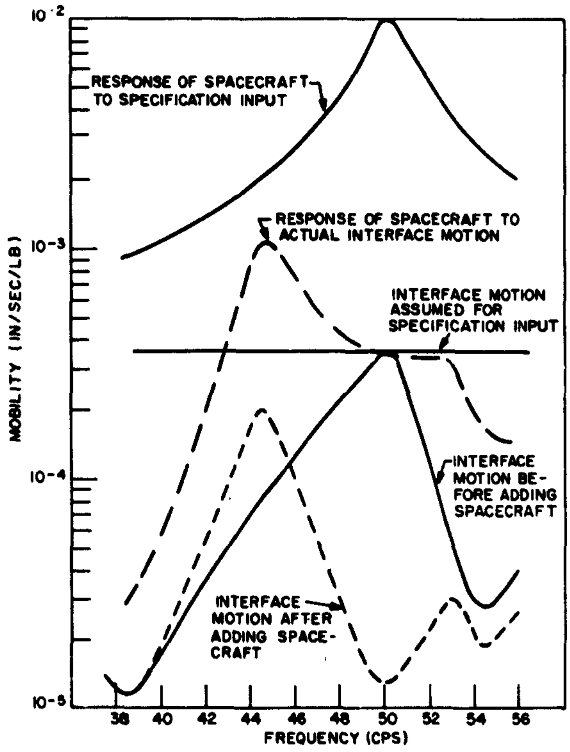

Fig. 5 - Spacecraft and launch vehicle response (mobility) curves for case of coincident spacecraft antiresonant and launch vehicle resonant frequencies

envelope of previously measured interface motions as "inputs" to the test specimen, since these motions are now part of the response of a new coupled system. Assuming broadband excitation, this response occurs primarily at the resonant frequencies of the coupled system, and not at either the resonant frequencies of the supporting structure alone or the antiresonant frequencies of the test specimen alone.

The use of a conventional motion input test results in the highest vibration levels occurring at the specimen antiresonant frequencies, whereas for specimens with significant impedance there is no reason to expect the highest levels to occur at these frequencies in the operational environment. Furthermore, a specimen with significant impedance loads down its supporting structure at its antiresonant frequencies and causes a reduction in the "input" motion. For the spacecraft-launch vehicle combination shown in Fig. 5 a reduction by a factor of over 25 was observed at the first spacecraft antiresonant frequency. No allowance is made for this effect in a conventional motion input test.

One modification of the motion input test that is commonly used permits a reduction in the input at critical antiresonant frequencies to avoid unrealistically high vibration levels or structural loads. An alternate and more straightforward approach is to abandon the motion input altogether, define instead an envelope of the anticipated response on the primary structure of the specimen in its operational environment, and use this envelope to specify the response on the primary structure of the specimen during its vibration test. It is important to point out that in using this method, the response cannot be defined for any single monitoring point, since at the antiresonant frequencies associated with the point chosen, large amplifications may occur on the test specimen in the same manner as they do at the driving point antiresonances during a motion input test. To avoid this condition, it is necessary to define the test envelope by the levels of several monitoring points distributed over the primary structure. The envelope may be defined as either the average vibration level of all the monitoring points, or the highest level shown by any single point. Reasonable arguments exist for using either of these approaches, however, it is not clear at present which is the most desirable. If the maximum level is used, care must be exercised to avoid having the test unduly influenced by very localized response. On the other hand, if the average level is used, the possibility exists of unrealistically high vibration levels over a small portion of the structure. In either case, the levels used should be representative of vibration on primary structure, and large amplifications occurring during the test on secondary structure should not be arbitrarily considered as grounds for reducing the overall test levels, since these amplifications may actually be representative of the operational environment.

For large impedance specimens the response envelope will probably not be appreciably above the envelope of measured data conventionally used as an input, and will not be subject to variations as large as those experienced by the interface in the frequency ranges near the specimen antiresonances. In the case of the spacecraft-launch vehicle combinations considered in Figs. 4 and 5 the maximum spacecraft response was approximately three times that observed at the 50-cps launch vehicle resonance before the spacecraft was attached. However, the response of the launch vehicle to the same excitation at some of its other resonant frequencies (e.g., 21 and 115 cps) would be even higher than the spacecraft response occurring in the coupled system in the 50-cps range, so that a broad flat envelope through the peaks in the launch vehicle response would also include the maximum spacecraft response for this critical frequency range. For this case, therefore, a vibration envelope at or slightly above the levels conventionally used as inputs would be suitable to define the spacecraft response.

Since only one case has been investigated to a limited extent in this paper, considerably more work is needed before general rules can be developed for establishing response envelopes for a large variety of specimens. Qualitatively, it is evident that as the specimen impedance becomes smaller the response envelope for testing must be moved further and further above the envelope of measured data until eventually it becomes more realistic to return to a motion input test. Until general rules can be developed it may be necessary to do simple impedance calculations of the type shown in this paper for each individual case before a conservative response envelope can be established with confidence.

The specification of response, rather than input, has several decided advantages over a conventional motion input test for specimens having significant impedance. First, it does not produce greatly increased response levels at the specimen antiresonant frequencies; rather it subjects the specimen to more nearly the same intensity of vibration throughout the frequency range, thus providing more equal assurance of acceptability for resonances occurring at any frequency. Second, by defining the

response, the degree of conservatism in the test becomes more predictable, since variations in peak response are less pronounced than variations in the "input." Finally, generally higher vibration levels can be used throughout the test range, since the difficulties associated with large antiresonant amplifications are eliminated. Some increased difficulty in instrumentation and shaker control will inevitably occur in using this test procedure, but this will not be serious enough to outweigh the benefits obtained from more realistic vibration tests.

It is recognized that this discussion leaves many questions unanswered, some of which must remain unanswered until new test techniques are actually tried in the laboratory. It is hoped that others in the field will be encouraged by this paper to present their own views and suggestions, and that ultimately, more realistic vibration test and design procedures for structures with appreciable impedance will result from these discussions.

* * *

Section 4
ISOLATOR DESIGN

SUSPENSION SYSTEM DESIGN TO REDUCE HIGH INTENSITY SHOCK*

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This paper presents basic concepts that must be recognized and used in the design of resilient suspensions to attenuate high intensity shock. The theoretical response of a given system may be calculated from measured test data for the Navy Hi-Impact Shock Testing Machine and known fragility data for the equipment to be protected. Such a design problem is reviewed. Actual test data taken during shock testing of the system is compared to the calculated response and proves the design approach.

INTRODUCTION

High intensity shock is a fundamental type of dynamic loading which frequently causes equipment malfunction. It is normally encountered in naval shipboard service as the result of explosive blasts or high velocity impacts. Frequently, however, it is used in the laboratory under more controlled conditions to determine equipment functional integrity. Even under controlled conditions, high intensity shock as a system input is difficult to consider analytically because of the complex relationship between parameters which may be used to describe it. This naturally makes difficult the task of designing equipment which will function properly after it is subjected to this environment.

Carefully engineered suspension systems may be used to attenuate high intensity shock to tolerable levels. Their use simplifies the overall design problem because response to the shock input is predictable. Substantial system deflection must be permitted, however, if the high intensity shock is to be attenuated appreciably. This cannot be obtained with hard mounted equipment installations.

Thus, the need to introduce additional suspension system elements, if high intensity shock attenuation is required, is evident. These elements, however, must be engineered to do a specific job. Introduction of suspension system elements with the wrong type of stiffness, damping, and deflection characteristics could cause more severe shock response than hard mounting.

TERMINOLOGY AND UNITS

$A_{M.R.}$ - Maximum response acceleration for shock inputs (g)

d - General symbol for deflection (inches)

*This paper was not presented at the Symposium.

$d_{M.B.}$ - Minimum possible deflection of an effective buckling isolator (inches)

$d_{M.L.}$ - Maximum required deflection of an effective linear isolator (inches)

d_{shock} - Simple system relative displacement due to a step velocity shock input (inches)

$d_{S.L.}$ - Deflection due to static load (inches)

$d_{V.L.}$ - Linear deflection required for vibration (inches)

E_{stored} - Energy stored in elastic member (inch-pound)

f_n - Simple system natural frequency (cps)

$f_{n(comb)}$ - Natural frequency of a simple system with combined structural and isolator stiffnesses (cps)

K - General symbol for spring rate (pounds/inch)

K_b - Supporting structure stiffness, spring rate (pounds/inch)

K_c - Combined isolator and structural stiffness (pounds/inch)

K_s - Suspension stiffness, spring rate (pounds/inch)

m_e - Mass of equipment (lb sec^2/inch)

P_{max} - Maximum force transmitted by isolator (pounds)

$P_{S.L.}$ - Static load (pounds)

ΔV - Step velocity shock input (inches/second)

$x(t)$, $\dot{x}(t)$ - Input excitation functions, displacement and velocity

$y(t)$, $\ddot{y}(t)$ - Response functions, displacement and acceleration

THE ENGINEERED SUSPENSION SYSTEM

The excitation from high impact testing or actual shock conditions has many frequency components and an actual electronic equipment has many normal modes of response dictated by the parts which make it an assembly. As a result, if such equipment is "hard mounted" and subjected to shock, these components will have a higher probability of failure or malfunction than would be possible if the unit were supported on a properly designed suspension. This is so because the components are subjected to the full intensity of the complex excitation in the former case and to a "filtered" excitation in the latter. Since it is impossible to design or modify a complex piece of equipment so that it does not have many normal modes of responses, and since these modes will in all probability be violently excited by shock when the unit is hard mounted, reliability under such conditions is difficult to obtain. To pass tests, modifications can be made to the equipment but, since equipment response under "hard mounted" conditions is nonlinear and complex, and therefore not predictable, it may be extremely fallacious to assume that this insures reliability under actual conditions.

On the other hand, the response of an equipment supported by a properly designed suspension is predictable for equal conditions and is therefore a significant help to the equipment designer responsible for reliability under dynamic loading. Once a given practical level of maximum transmitted acceleration is decided upon, the characteristics of the suspension necessary to assure this can be determined. Then all further equipment mechanical design can be continued with a higher degree of confidence that future shock tests will be successfully passed, and necessary modifications minimized. More important, the response to actual conditions is more likely to be close to that which was anticipated by analysis. This is the strongest single argument for use of properly designed shipboard electronic equipment suspension systems.

SYSTEM SHOCK RESPONSE

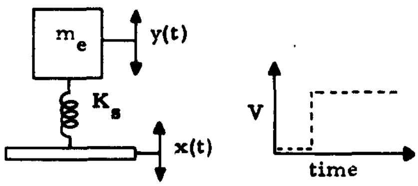
The true nature of shock expected aboard operational ships is now being studied in detail. The Navy Hi-Impact Shock Test has been used to determine equipment suitability, since early World War II. The damage, failures, and malfunctions resulting from this test are similar to those resulting from battle conditions in operational service. Indeed, the test was contrived for the purpose of reproducing in the laboratory, the observed effects of shipboard shock.

In contrast to actual shipboard shock as encountered in operation, the shock characteristics

of the Hi-Impact Test Machines have been thoroughly studied and well documented [1,2].

Taking these into account, shock as produced by the Hi-Impact Test Machines may be considered as a step velocity change of 10 to 12 fps for purposes of determining maximum values of simple system response. Shock spectra data correlate very well with response analysis of the simple systems which treat the shock excitation as a 10- to 12-fps step input. In addition, actual measurements of the impulsively attained velocity of the test machine anvil table indicate this value [3].

The theory of how a simple system responds to a step input velocity change has been adequately presented in several references [4,5]. However, the development of this theory and the resultant conclusions depend upon the use of differential equations or operational mathematics and, therefore, is a confusing subject to many engineers. The conclusions are of principal interest and are summarized here to support later recommendations. Generally, the important factors of simple system response to shock are the maximum acceleration transmitted to the mounted equipment and the displacement across elastic members in the system (relative motion between equipment and support). Note that these are the same factors that are important in determining response to vibration inputs.



Note: The effect of damping on maximum values of shock response is negligible for the system being considered.

Fig. 1 - Simple model with step velocity input

Specifically, for a step velocity input to the simple system shown in Fig. 1, the maximum response acceleration [4] is

$$A_{M.R.} = \ddot{y}(t) = \frac{(\Delta V)\, f_n}{61.4}\ g, \qquad (1)$$

where:

$A_{M.R.} = \ddot{y}(t)$ = maximum acceleration of equipment (m_e) g,

$\Delta V = \dot{x}(t)$ = magnitude of input step velocity change in inches/second, and

f_n = system natural frequency in cps

$= \frac{1}{2\pi}\sqrt{K_s/m_e}$.

Note in Eq. (1) that as suspension natural frequency is decreased, the maximum equipment acceleration is also decreased for a given step velocity input.

The displacement across the elastic member (K_s) of the suspension is given by the simple expression:

$$d_{shock} = \left[x(t) - y(t)\right] = \frac{23}{f_n}\ (\text{for } \Delta V = 12 \text{ fps}) \qquad (2)$$

or

$$d_{shock} = \frac{19.2}{f_n}\ (\text{for } \Delta V = 10 \text{ fps}), \qquad (3)$$

where:

d_{shock} = displacement across K_s in inches, and

f_n = system natural frequency in cps.

Now note that if suspension natural frequency is to be decreased, isolator linear deflection must increase. Tieing this in with Eq. (1) leads to the conclusion that to reduce transmitted acceleration to a mounted equipment, the deflection of the suspension must increase for a given system input. Equations (1), (2), and (3) have been plotted in Fig. 2 for a step velocity change of 10 to 12 fps.

The foregoing is true for a suspension system with ideal linear elasticity. Such a

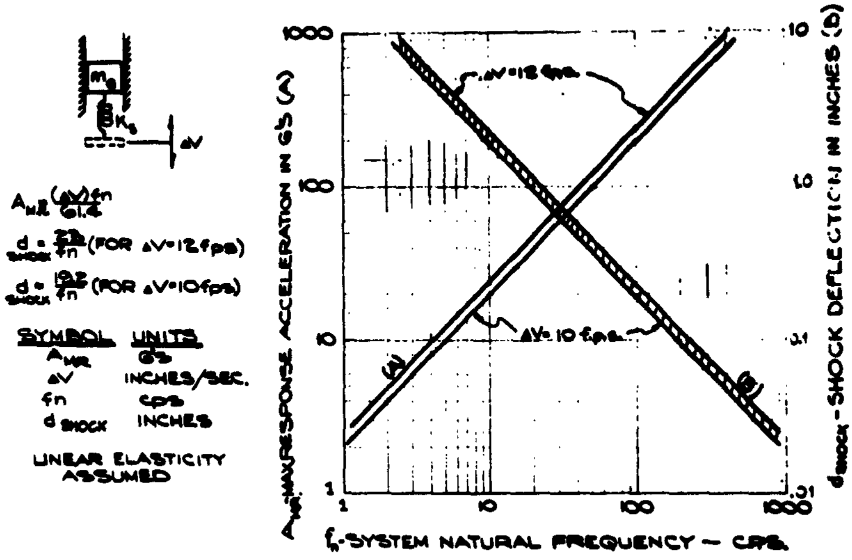


Fig. 2 - Maximum response acceleration and relative deflection as a function of system natural frequency for an undamped single-degree-of-freedom system subjected to a step velocity input

system has a load-deflection characteristic as shown in Fig. 3(a). The stiffness, or spring rate (K), is defined as the slope of the load-deflection curve and is the same value at all deflections. Actual isolators can be designed to have approximately linear stiffness.

Fig. 3(b) shows the load-deflection curve of an elastic element with exponential stiffness. For this element, the stiffness varies as a function of deflection and, in Fig. 3(b), increases with deflection. This type of load-deflection characteristic is common in many commercially available isolators. In the extreme case, the shape of the load-deflection curve is like that of Fig. 3(c) where the final stiffness (K_1) is many times greater than the initial slope (K_0).

The ability of an elastic member to store energy depends upon the shape, and is the integral of, the load-deflection curve. Thus, the area under the curve at any deflection is equal to the stored energy and, for example, in the case of the linear element (Fig. 3(a)) is

$$E_{stored} = 1/2\ Kd^2 \text{ (linear element)}. \qquad (4)$$

Note that a linear element will store more energy for a given amount of deflection and a given maximum load than will elements as shown in Figs. 3(b) and 3(c). Furthermore, if energy storage was the only criterion for an element that must deflect, the "square-wave absorber" shown in Fig. 3(d) would be most effective because it requires a minimum amount of deflection to absorb a given amount of energy. It is twice as efficient as the linear element. Unfortunately, it is not possible to design a practical suspension system element with ideal "square-wave" characteristics. However, parts with load-deflection characteristics approximating those of a square-wave element are feasible, but increasingly difficult to design as the "ideal" is approached, see Fig. 3(e). These elements are commonly called "buckling mounts."

Now note that for a given amount of energy storage (area under the load-deflection curve), the elements of Figs. 3(b) and 3(c) required much greater maximum force (P_1) than the linear or buckling type element. Since the maximum force in the elastic element is directly proportional to the maximum acceleration transmitted to the equipment lump mass (m_e), suspension elements like those of Figs. 3(b) and 3(c) are not suitable at all for shock isolators where the objective is to minimize transmitted forces and isolator deflection. It is true that exponential stiffness elements can be designed to keep deflections within any bounds. But what a price is paid in transmitted forces! Furthermore, the natural frequencies of these elements vary as they are deflected and thus they pass the complex shock excitation at higher amplifications than linear or buckling elements.

Another extremely important consideration in the design of shipboard equipment suspension

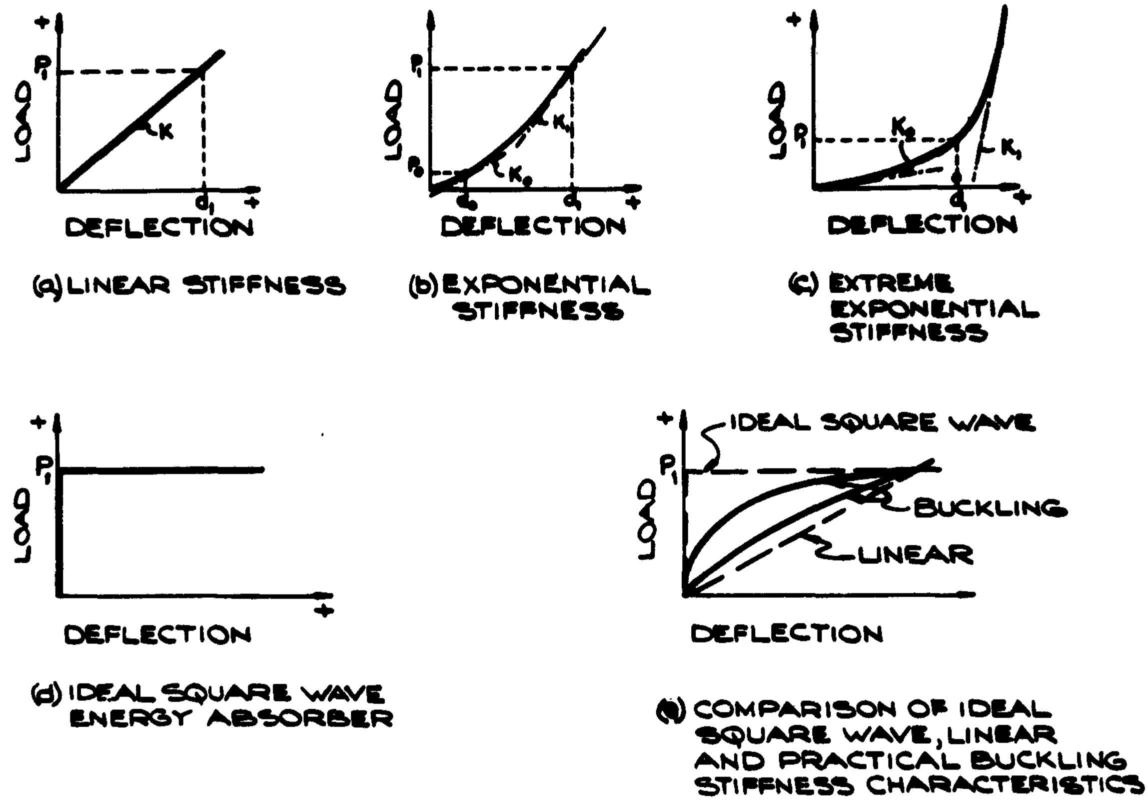


Fig. 3 - A comparison of the load-deflection characteristics of various elastic elements

systems is installation geometry. This consideration cannot be overemphasized. As an example, improper system configurations were observed in the majority of equipment installations aboard a typical fleet destroyer in a ship-by-ship survey [6]. This points out the necessity of considering installation geometry as a principal factor in systems design.

For shipboard installations, where space is almost always at a premium, a center-of-gravity (cg) mounting arrangement should be used. This may be achieved by using either of two methods (Fig. 4):

- The Perfect cg Support — where the isolators lie in a plane that is parallel to the principal geometric axes of the equipment and contains the equipment center of gravity; or

- The Diagonal cg Support (also referred to as stabilized base support) — where the isolators lie in a plane that is skew to the principal geometric axes of the equipment and contains the equipment center of gravity.

The advantages of such an installation configuration are:

- Equipment sway space is reduced to that required for isolator travel because coupled (rocking) modes of the equipment are eliminated.

- The number of different configurations (load rating) of isolators to achieve decoupled response usually is minimized. This means attachment bolt locations and hole plans at each isolator can probably be the same.

- Axial and radial isolator stiffness characteristics can be the same for units that have an approximately symmetrical center-of-gravity location. This has had the added advantage of allowing flexibility in selecting the normal attitude or position of isolators at each location.

- Response acceleration levels at each point of attachment on the equipment will be approximately the same because rotational modes due to translation inputs have been

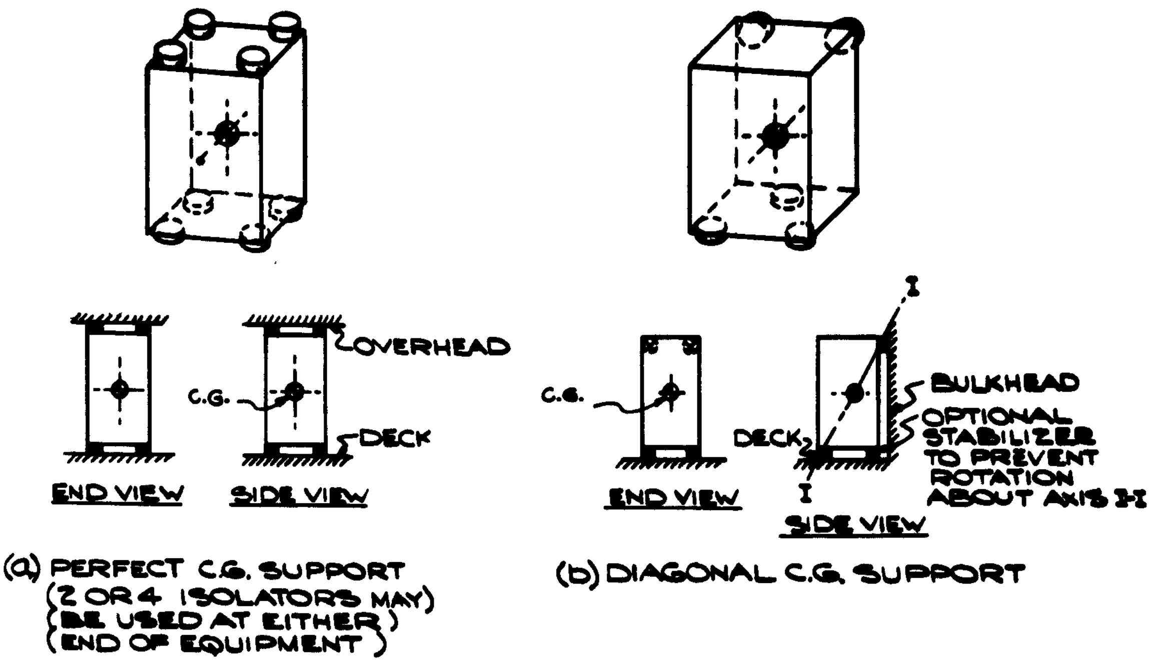


Fig. 4 - Recommended suspension system geometry (schematic) for shipboard equipment installations

eliminated. This simplifies the loading considerations in equipment mechanical design.

• Various equipments may be mounted on a common frame and this "lump-system" supported on suitable isolators. This allows a "modular" packaging approach which may be very effective in utilizing space.

Another very important consideration in the installation of such systems is the stiffness of supporting and equipment structures. The structural stiffness should be at least five times the stiffness of the suspension system.

For example, consider the simple system (Fig. 5). If the mounted equipment (m_e) on its suspension system (K_s) has a natural frequency of 15 cps and the independent stiffness of the support K_b is only twice that of the system, K_s, that is

$$f_n = \frac{1}{2\pi}\sqrt{K_s/m_e} = 15 \text{ cps}, \tag{5}$$

and

$$K_b = 2K_s, \tag{6}$$

then the series stiffness combination of support K_b and K_s is

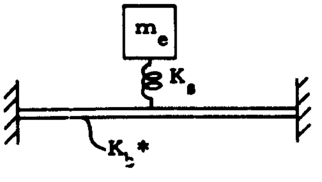
$$K_c = \frac{K_s K_b}{K_s + K_b} = \frac{2K_s^2}{3K_s} = \frac{2}{3}K_s. \tag{7}$$

Put another way, the effect of the series stiffness combination on the system natural frequency (Eq. (5)) is to reduce the natural frequency,

$$f_{n(comb)} = 2/3 \quad f_n = 12 \text{ cps}.$$

Thus, system natural frequency has been reduced 20 percent because of a relatively "soft" supporting structure. If the ratio K_b/K_s had, in comparison, been the recommended minimum of 5, then

$$f_{n(comb)} = 14 \text{ cps},$$



*Mass effect of support not considered.

Fig. 5 - Simple system and support

or a change of only 6.7 percent. Furthermore, deflection of the supporting structure under high levels of dynamic loading would only be 2/5 that of the system in which $K_b = 2K_s$.

A good general design practice is to attach mountings to foundations or structural stiffeners rather than to panels, bulkheads, or decks [7].

The foregoing comments summarize the principal factors to be considered when planning suspension system installations. This subject is covered in great depth in chapters 1, 2, and 3 of Ref. 8.

SUMMARY OF SUSPENSION CHARACTERISTICS

It can therefore be concluded that an effective shipboard shock isolation system will have the following characteristics:

- Linear or buckling stiffness with sufficient travel to accommodate the deflection required for shock protection as indicated in Fig. 1;
- Linear deflection only up to the maximum required for static load (if applicable depending on direction of static load) plus vibratory excursions;
- Gradual snubbing for excursions beyond those anticipated due to shock, with no hard or uncushioned bottoming permissible; and
- Captive assembly incorporated into the isolator assembly or used separately in parallel with the isolator. Not to take effect except at excursions equal to or greater than the maximum anticipated during shock.

System damping of practical magnitudes has only a slight influence on shock response. If, however, vibration requirements indicate it is necessary, it should be incorporated into the system elements for that purpose.

Figure 6 shows the stiffness characteristics of isolators suitable for shipboard electronics equipment suspensions. All useable designs will have characteristics within the upper and lower boundaries. Definitions of the quantities are as follows:

$d_{M.L.}$ - The maximum deflection of a linear isolator. It is determined from Fig. 2 by establishing the system natural frequency and then finding the shock deflection. Add 1/8 inch minimum to this shock deflection to allow for gradual snubbing.

$d_{M.B.}$ - The minimum possible deflection of a practical buckling isolator. $d_{M.B.}$ equals approximately 2/3 $d_{M.L.}$

$d_{S.L.}$ - Isolator deflection of static load. $d_{S.L.} = 9.8/f_n^2$ inches.

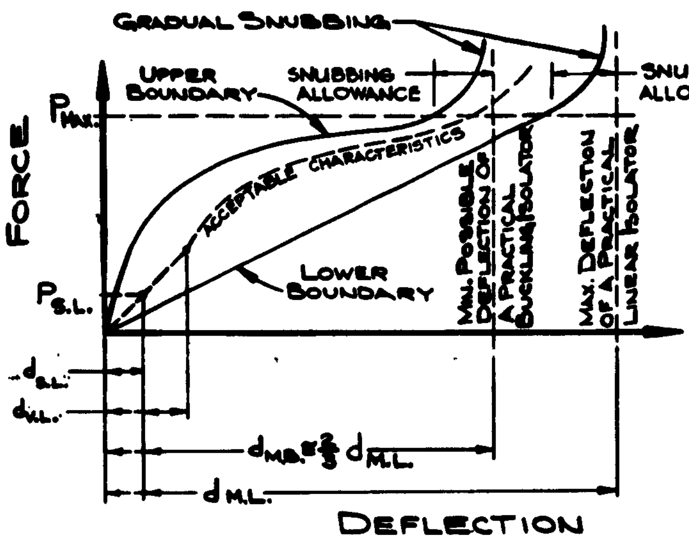


Fig. 6 - General characteristics of an effective shipboard electronics suspension system isolator

$d_{V.L.}$ - The linear deflection required for vibration conditions. It may include an allowance for static deflection depending upon isolator orientation.

P_{max} - Maximum force transmitted to support equipment attachment points under shock conditions.

$P_{max} = (P_{S.L.}) \times (A_{M.R.})$

$P_{S.L.}$ - Static load

$A_{M.R.}$ - Maximum response acceleration determined from Fig. 2.

PROTECTING A SHIPBOARD INERTIAL PLATFORM

An inertial platform was intended for use aboard naval vessels. To meet naval requirements however the platform had to be subjected to high impact shock testing in accordance with MIL-S-901B. The fragility limit of the unit was established as 15-g maximum acceleration. That is, accelerations in excess of this value would cause equipment malfunction.

Shock inputs from the Navy Hi-Impact Shock Machine for lightweight equipment vary from 100 to 1000 g, depending on the frequency spectrum considered. Obviously, the inertial platform, if hard mounted, could not survive such shock inputs.

In addition to the shock requirements, the unit had to be subjected to vibration tests in accordance with MIL-STD-167 (Ships). For this requirement, platform-servo natural frequencies dictated a suspension system natural frequency of less than 8 cps. This worked out very nicely with the system natural frequency required to meet the shock requirements.

A simple system analysis of this problem, using the methods outlined earlier in this paper, indicated that a system natural frequency of approximately 7 cps, or less, would be required to attenuate the Hi-Impact Shock Test input to a maximum platform acceleration of 15 g or less (Fig. 2). In addition, for a linear system the response deflection had to be at least 3 inches.

Another system requirement was that the angular position of the inertial platform relative to the ship's supporting structure, be maintained within 1 minute of arc during vibration and after subjection to shock. This requirement arose from the fact that the inertial platform is used as a directional reference and that the angular alignment relative to the ship's structure must be maintained if the equipment is to be effective. Such a requirement might indicate that "hard mounting" is the only solution to maintain such angular positional tolerance. However, the fragility limit of 15-g maximum acceleration could not be ignored.

A suspension system was devised to satisfy all of these requirements. In general, it closely resembles the simple system examined earlier in this report; the characteristics recommended in this article were incorporated in its design. Figure 7 shows the suspension system completely assembled with the platform installed.

The platform is supported by a cradle assembly fabricated from aluminum castings. Chrome-plated hardened-steel tubes pass through bearing housings at the ends of the cradle and allow vertical translation only of the platform and cradle relative to the supporting structure. A similar tube-bearing arrangement is provided for lateral and longitudinal translational freedom. These "gimbal" stages maintain true rectilinear motion, provide high angular restraint, and still allow the system to deflect the 3 inches minimum required for attenuation of shock inputs. Natural frequency of each gimbal stage of the system from the "outside-in" was 5, 6, and 8 cps. Helical springs between numbers having relative motion in each gimbal stage provide the required system linear elastic characteristics. Step friction dampers incorporated into the bearing housings at the various gimbal support points keep resonant vibration transmissibility within reasonable limits without degrading isolation of exciting frequencies above 11 cps. Recirculating ball bushings incorporated into each of the gimbal support points and adjusted to eliminate all mechanical backlash provide virtually frictionless guidance and translational freedom.

Gimbal stages are supported by a double "A" frame fabricated from aluminum tubing. This frame, in turn, is secured to a hollow circular base which has very high rigidity although it is light in weight. Welding is used throughout to fasten together the supporting structural members.

A simple system of cables and pulleys act as a rigid structure between the upper and lower crossover fittings (Fig. 7). This arrangement is necessary to eliminate relative motion between these members. The complete assembly with dust cover is 25 inches in diameter by 30 inches high.

Fig. 7 - Inertial platform isolation system with platform installed

Note that a minimum amount of space has been used for supporting structure although structural members must be extremely rigid to withstand the high intensity shock inputs to which they must be subjected. The complete unit with an inertial platform dummy load and dust cover installed is shown in Fig. 8. Note that an access panel has been provided at each end of the dust cover. These panels allow longitudinal motion of the inner gimbal assembly. Under shock conditions, the platform, its support cradle, and the inner gimbal may deflect up to 4-1/2 inches in the fore and aft direction. This action pushes the access panels away from the dust cover thereby allowing the translational freedom required to attenuate shock. The panels follow the motion of the gimbal and are returned to their original positions by elastic restraining cords after the shock is over. Such a feature minimizes the overall size of the installed assembly.

At first glance, it may seem that this system is complex; however, it should be kept in mind that it must:

- Attenuate the Navy Hi-Impact Shock Test accelerations from 100 - 1000 g to 15-g maximum,

- Provide protection from MIL-STD-167 (Ships) vibration inputs and have a natural frequency of approximately 8-cps maximum, and

- Maintain angular alignment of the inertial platform relative to the ship's supporting structure within 1 minute of arc during vibration and after shock. (For comparison purposes, 1 minute of arc represents 0.0003 inch per inch deviation.)

TEST RESULTS

A prototype system was constructed and tested in accordance with the requirements of MIL-S-901B for lightweight equipment. A typical test installation is shown in Fig. 9. The test requires shock inputs in all three coordinate directions of the suspension system. Calculated and actual test results are shown in Table 1 for comparison purposes.

Fig. 8 - Inertial platform isolation system with dust cover installed and aft access panel removed

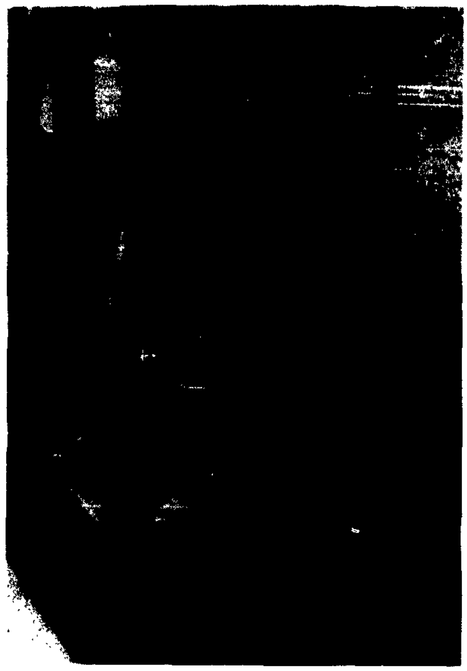


Fig. 9 - Inertial platform suspension system installed on High-Impact Shock Testing Machine for lightweight equipment and oriented for a side blow

TABLE 1

Summary of Calculated and Measured Shock Response of Inertial Platform Suspension System [9]

Direction of Shock Impact	Hammer Height (ft)	Measured Peak Response[a] Acceleration (g)	Calculated Peak Response[b] Acceleration
Vertical Blow	1	0.5	6.2
Vertical Blow	3	2.7	10.2
Vertical Blow	5	5.1	12.5
Side Blow	1	1.7	6.9
Side Blow	3	4.8	11.7
Side Blow	5	7.5	11.7
Back Blow	1	7.5	7.0
Back Blow	3	7.5	10.6
Back Blow	5	10.6	12.7

[a]Response acceleration measured on dummy inertial platform.

[b]Calculated response using Eq. (1) and system natural frequencies of $f_{n\,vert}$ = 8 cps, $f_{n\,Lat\,(Side)}$ = 5 cps, $f_{n\,Long\,(Back)}$ = 6 cps, and step velocity input data as presented in Ref. 1, pages

Note that the suspension system provided the required protection for the inertial platform by a reasonable margin. The difference between measured and calculated values of acceleration may be accounted for by the fact that the test machine was used at maximum load capacity thereby reducing the intensity of the test input to less than the values used in the calculations. Actual test inputs were not measured, although mockup response was. The oscilloscope trace for the 5-foot vertical blow is shown in Fig. 10.

Vibration tests in accordance with MIL-STD-167 indicate that the system does indeed have natural frequencies in the three coordinate directions of 5, 6, and 8 cps. As can be seen by examining the transmissibility curves in Fig. 11, damping was sufficient to limit resonant transmissibility to 2 or less.

SUMMARY

The fundamental aspects of the problem of determining shock isolation system characteristics for shipboard equipment have been covered in detail. A design philosophy for such isolation systems has been proposed and its value proved by the application reviewed.

It is entirely possible to design isolation systems to meet stringent shock attenuation requirements. To do this, however, sufficient system deflection must be allowed and can be justified if the shock isolation problem is considered from the beginning of equipment design. The excellent references listed at the end of this paper should be consulted to examine the subject in more depth. They represent the combined efforts and experiences of qualified experts in this field of study. A wealth of related information will be obtained from them by the interested reader.

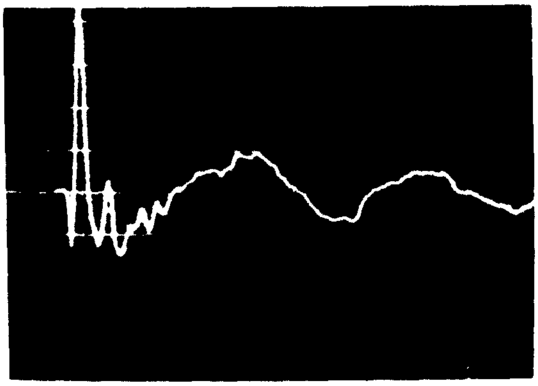


Fig. 10 - Oscilloscope trace of acceleration transmitted to isolated inertial platform mockup. Vertical Blow - Five feet (No. 1). Horizontal sweep - 50 ms/division; vertical gain - 100 mv/division; calibration - 91.5 mv/g, and acceleration level - 5.0 g.

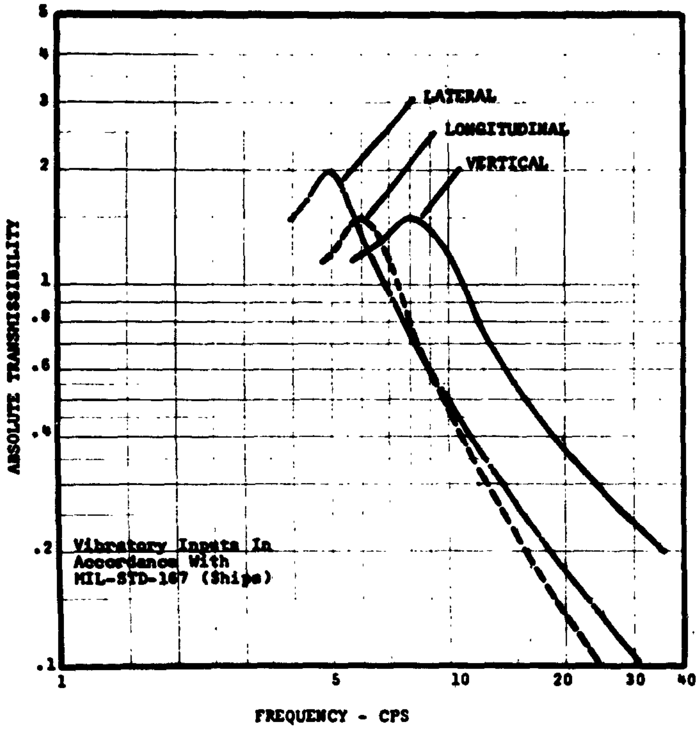


Fig. 11 - Vibration response of inertial platform isolation system to MIL-STD-167 (Ships) inputs

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* * *

USE OF THE ANALOG COMPUTER TO STUDY CUSHION CHARACTERISTICS AND PACKAGE DESIGN*

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This paper describes the force analyzer and the analog computer which have been designed specifically for package and cushion analysis.

INTRODUCTION

The Air Force Packaging Laboratory located at Brookley Air Force Base, Mobile, Alabama, has capabilities for testing, analyzing, and performing research relating to packaging design. The Laboratory is also equipped to test materials for mechanical characteristics as well as quantitative and qualitative composition. The Laboratory is equipped to simulate environmental conditions as they exist throughout the world. Some space environments also may be duplicated.

This paper deals specifically with the force analyzer and the analog computer, which together constitute an elaborate electronic installation designed especially for package and cushion analyses. This installation is housed in a centrally-located, environmentally-controlled room and is the receiving point for signals from all apparatus located throughout the laboratory. The analyzer console houses a bank of 48 Philbrick amplifiers, each multiplying the incoming signal by a factor of 10. Since these multipliers are between the apparatus input and the recorder input, compensation for line loss can be made. The Philbrick amplifiers have a differential capability, that is, they tend to cancel out all extraneous signals other than the desired data signal, which in most cases comes from an accelerometer set. The amplified data may then be directed to a memoscope, an 8-channel recorder, a tape record, a two channel x, y recorder, or the analog computer. Visual observation of a process is monitored by closed circuit television (Fig. 1).

*This paper was not presented at the Symposium.

The computer used for design analyses (Fig. 2) has the following features:

- 32 amplifiers (30 computing and 2 auxiliary)
- 60 coefficient helipots
- 18 helipots which set initial conditions
- 6 function generators
- rotary commutator-to-scan amplifiers, for determining out of balance condition
- VTVM readout
- X, Y, Y_2 recorder
- AMP shielded patchboard with color coding.

PROJECT

It appears, from analysis of past data on package drops and dynamic drop tests of cushioning materials, that a damping coefficient (viscous damping, usually denoted as R) and a stiffness coefficient (usually denoted as K) would be sufficient to describe a given cushioning material. The forces acting on a simple cushion system are shown in Fig. 3. The general equation of motion for a cushioned article may be written as:

$$m \frac{d^2x}{dt^2} + F_d + F_s + mg = 0 ,$$

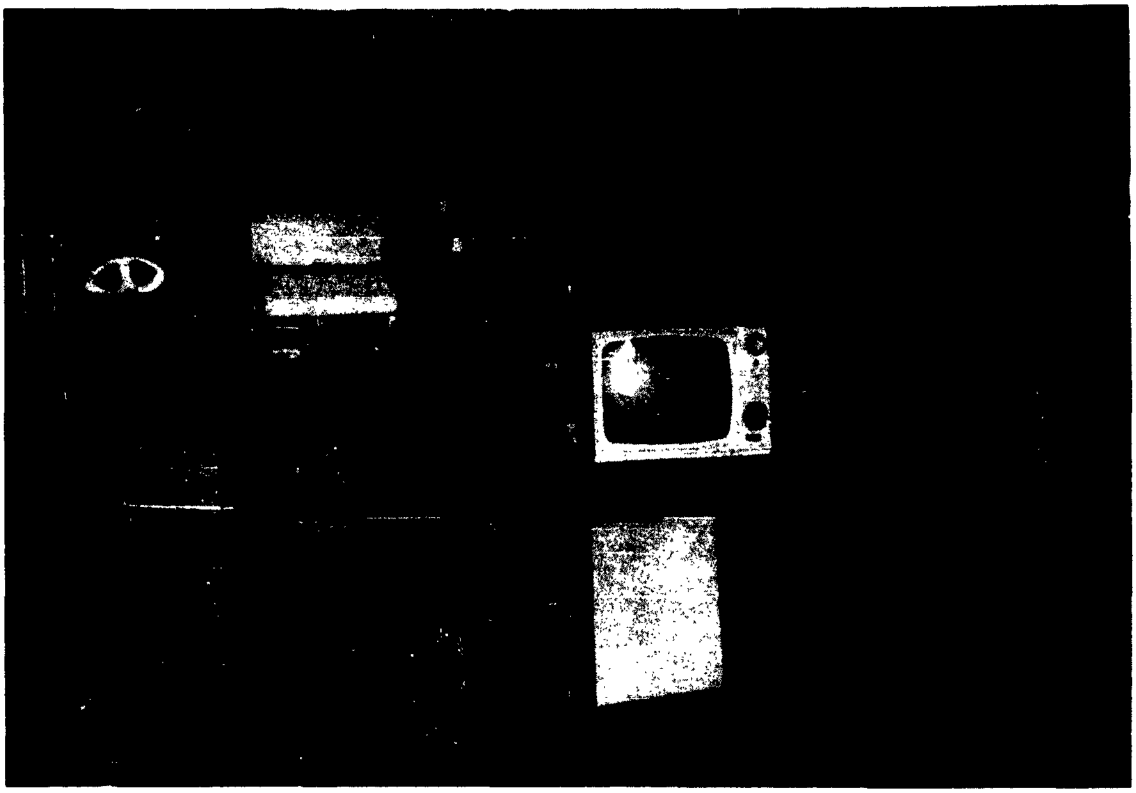


Fig. 1 - Force analyzer console, television monitor, and tape recorder

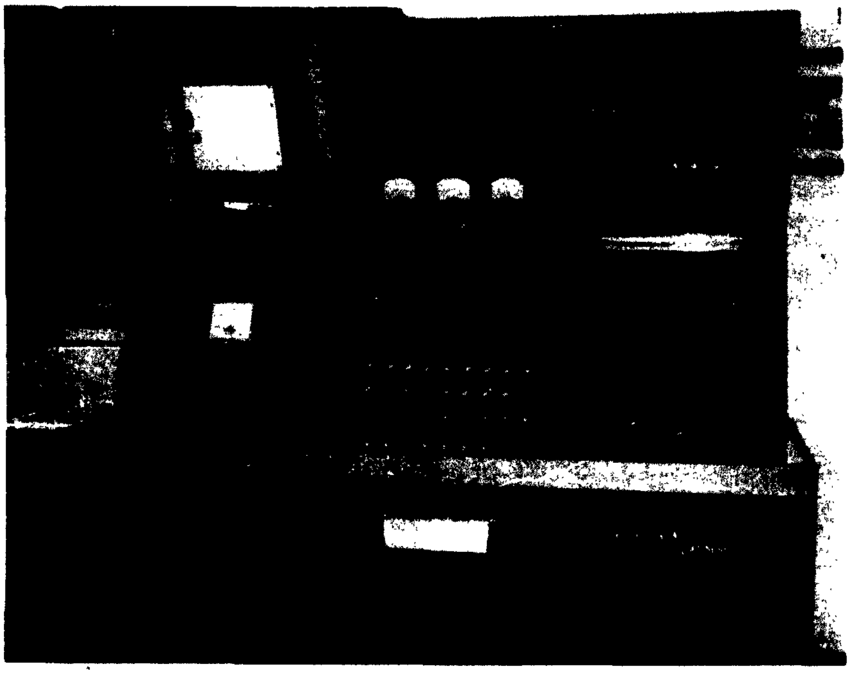


Fig. 2 - Analog computer

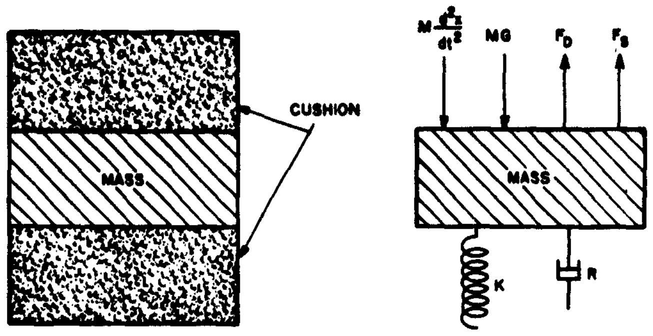


Fig. 3 - Forces acting on a simple cushion system

where

m = mass of article,

$\frac{d^2x}{dt^2}$ = instantaneous de-acceleration,

g = acceleration due to gravity,

F_d = force due to damping of cushion, and

F_s = force due to stiffness of cushion.

Initial condition at $t = 0,\ x = 0$;

$$\frac{dx}{dt} = \sqrt{2gh}\ ,\quad \frac{d^2x}{dt^2} = 0\ .$$

F_d will have the form $R(dx/dt)$, F_s will be

$$F_s = \frac{2K_0\, d_b}{\pi} \tan \frac{\pi x}{2d_b}$$

or

$$F_s = K_0 x + rx^3\ ,$$

where

K_0 = initial spring rate,

d_b = maximum available displacement,

r = rate of increase or decrease of spring rate.

The equations for F_s are characteristic of most cushion materials and were taken from R. D. Mindlin's paper "Dynamics of Package Cushioning" (Fig. 4). The project of obtaining R and K for all standard cushion materials will begin in this Laboratory in the near future and will be based on the following theory: K and R for a given cushion material are functions of the supported mass (m), the bearing area (A), and thickness (t) of the cushion. We don't know these relationships yet, and no doubt other variables will enter into a practical package design, factors such as a compressed cushion inside an outer container, temperature pressure, humidity, and so on. For simplicity, the latter variables will be omitted. If the stated relationships between m, A, t, can be found and substantiated, there is no doubt they would be of great benefit to the field of packaging.

Let us assume that for a given cushion material, say rubberized hair, the relationships shown in Fig. 5a have been found empirically (these curves are purely hypothetical). The package engineer desiring to protect an item of given mass to a given g value could examine the curve corresponding closely to the mass and quickly note the stiffness (K) of cushion required to give this g value. He could then examine Fig. 5b to find the bearing area and thickness required to give the required K factor. This curve is for one cushion material. Perhaps the curve for a similar cushion material would give a more desirable thickness and bearing area for the required K factor. An examination of the curve in Fig. 5c reveals that for a given mass and stiffness the system will have a given damping coefficient. By using the equation in Fig. 5c, calculations may be made to find the natural frequency of the system. In practice, the data will be in table form rather than graphic form. The advantage in using a system such as this is that the information is all together in simple form. It contains all the technical information needed to design the cushioning system and package dimensions. It provides a method to calculate natural frequency and damping effects. To the best of our knowledge no data has been published on the latter.

The method to be used to find R and K values will be as follows. The elevator of a dynamic drop tester will be lifted to a height of

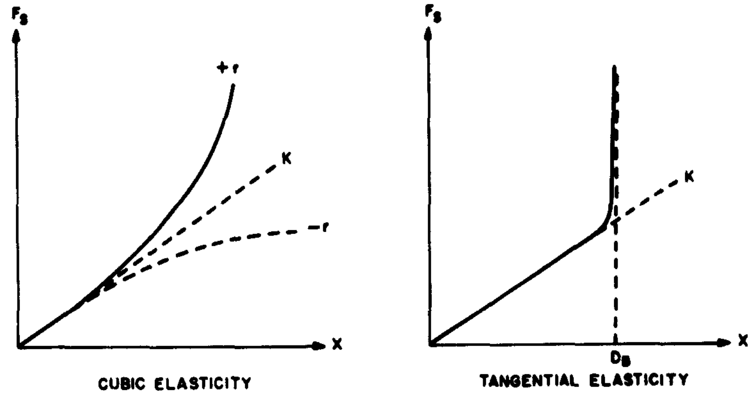


Fig. 4 - Mindlin's curves

30 inches and dropped on a given cushion. Output from an accelerometer mounted on the drop head will be amplified and recorded on tape. The tape output will then be played back (repeatedly) through the computer for analysis as discussed later. The general equation:

$$m \frac{d^2x}{dt^2} + R \frac{dx}{dt} + (K_0 x + rx^3) + mg = 0 ,$$

will be set up on the computer (Fig. 6). The curve for acceleration will come out of the computer at a repetitive rate. The repeating tape will then be synchronized with the computer. The computer curve will be compared with the data curve within a summing amplifier. We may now vary R and K on the computer until the two curves are identical. At this point one signal nulls the other and no signal is viewed on the oscilloscope. R and K may now be read directly from the helipot dials (Fig. 7). Curves similar to those of Fig. 5 will then be plotted by finding the R and K values for different drop heads, masses, cushion areas, and thicknesses.

In the computer we have the capability to analyze a complete package design by setting up the desired design parameters on computer

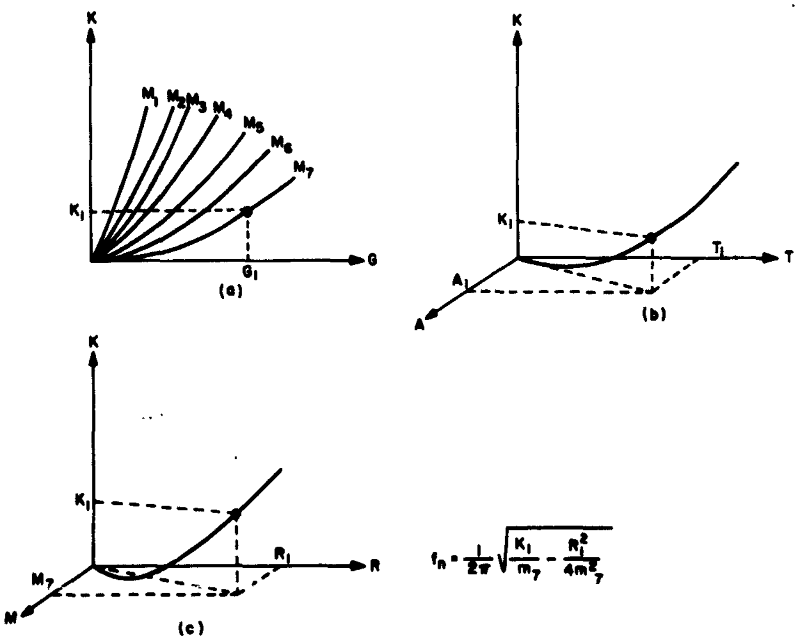
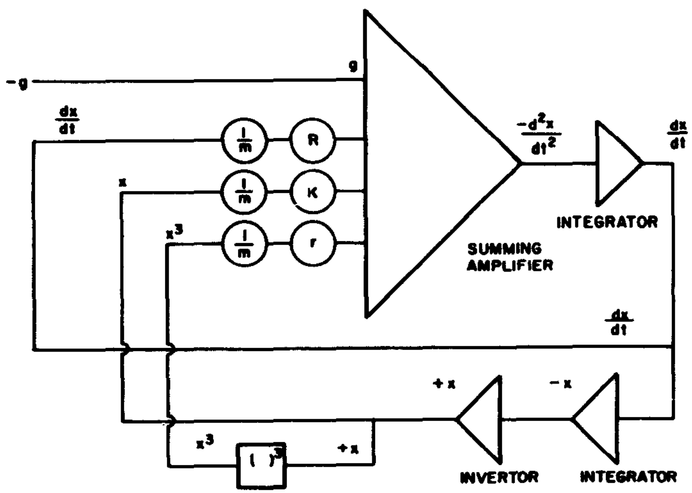


Fig. 5 - Procedure for selecting a package cushioning system



$$\frac{d^2x}{dt^2} = \frac{R}{m}\frac{dx}{dt} + \frac{K}{m}x + rx^3 - g$$

Fig. 6 - Computer block diagram

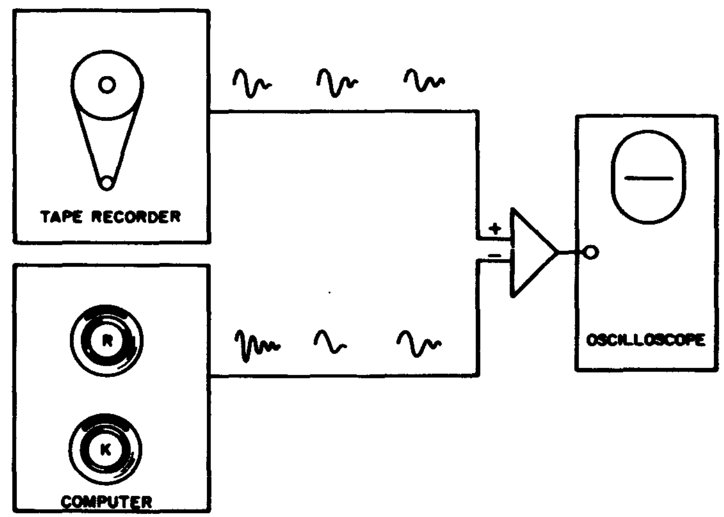


Fig. 7 - Signal comparison

helipots. That is, once we have found how R and K vary with mass of inner article, bearing area, and thickness of cushion, we may describe a package of given material by its dimensions. The design may then be subjected to any environmental condition also created within the computer by the function generators. Voids as well as blocking and bracing also may be simulated in the function generators.

In the computer, component motions as well as the resultants can be investigated for both the inner and outer container simultaneously. Figure 7 shows the designation of variables and parameters for a package with six degrees of freedom; three degrees for the inner package (x_1, y_1, and z_1) and three for the outer package (x_2, y_2, and z_2).

The equations of motion then take the outer container and its characteristics into consideration (Appendix A).

Figure 8 is a block diagram of the Y-component computer circuits for a six-degree package system.

Since our equipment has just been completed, we have been unable to obtain data. The project of determining cushioning characteristics and package design parameters will be built partially around the theories included in this paper. There are many points that will need justification and, with a view toward sophistication, more variables must be taken into account in the processes.

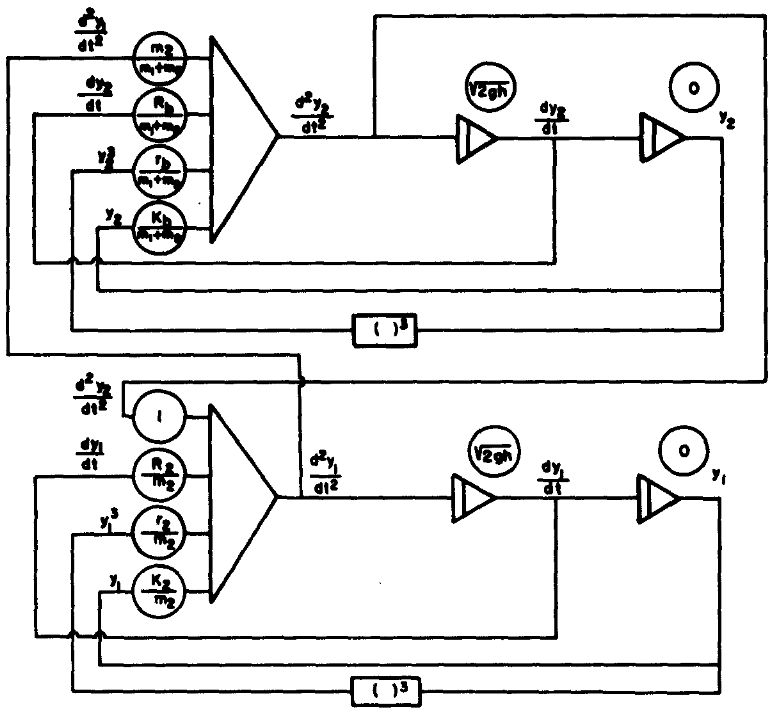


Fig. 8 - Y-component computer circuits for a 6-degree package system

Appendix A

EQUATIONS OF MOTION FOR PACKAGED ITEM IN CONTAINER

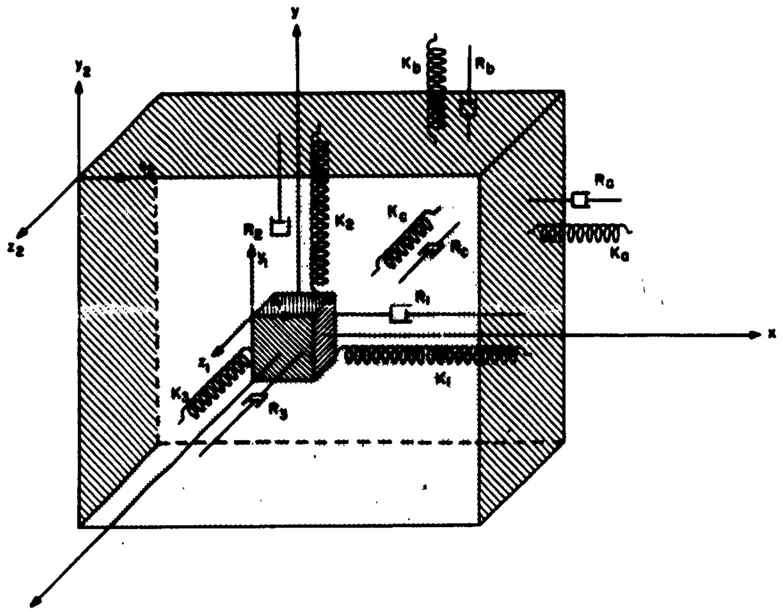


Fig. A1 - Schematic of packaged article, designation of variables and parameters

NOTATION

g = acceleration due to gravity

h = drop height

In the simple case (Fig. A1),

$$R_1 = R_2 = R_3$$
$$K_1 = K_2 = K_3$$
$$r_1 = r_2 = r_3$$
$$R_a = R_b = R_c$$
$$K_a = K_b = K_c$$
$$r_a = r_b = r_c$$
$$\alpha = 0°$$
$$\beta = 90°$$
$$\gamma = 0°$$

for flat drop.

EQUATIONS FOR INNER CONTAINER (FIG. A1)

$$m_1\left(\frac{d^2x_1}{dt^2} + \frac{d^2x_2}{dt^2}\right) + R_1\frac{dx_1}{dt^2}$$
$$+ K_1x_1 + v_1x_1^3 = 0,$$

$$m_1\left(\frac{d^2y_1}{dt^2} + \frac{d^2y_2}{dt^2}\right) + R_2\frac{dy_1}{dt^2}$$
$$+ K_2y_1 + r_2y_1^3 + mg = 0,$$

$$m_1\left(\frac{d^2z_1}{dt^2} + \frac{d^2z_2}{dt^2}\right) + R_3\frac{dz_1}{dt^2}$$
$$+ K_3z_1 + r_3z_1^3 = 0.$$

EQUATIONS FOR OUTER CONTAINER (FIG. A1)

$$(m_1 + m_2)\frac{d^2x_2}{dt^2} + m_1\frac{d^2x_1}{dt^2} + R_a\frac{dx_1}{dt} + K_a x_2 + r_a x_2^3 = 0,$$

$$(m_1 + m_2)\frac{d^2y_2}{dt^2} + m_1\frac{d^2y_1}{dt^2} + R_b\frac{dy_2}{dt} + K_b y_2 + r_b y_2^3 + mg = 0,$$

$$(m_1 + m_2)\frac{d^2z_2}{dt^2} + m_1\frac{d^2z_1}{dt^2} + R_c\frac{dz_2}{dt} + K_c z_2 + r_c z_2^3 = 0.$$

Initial conditions at $t = 0$

$$\frac{d^2x_1}{dt^2} = \frac{d^2x_2}{dt^2} = \frac{d^2y_1}{dt^2} = \frac{d^2y_2}{dt^2} = \frac{d^2z_1}{dt^2} = \frac{d^2z_2}{dt^2} = \frac{dx_1}{dt} = \frac{dx_2}{dt} = \frac{dz_1}{dt} = \frac{dz_2}{dt} = x_1 = x_2 = y_2 = z_1 = z_2 = 0,$$

or

$$\frac{dx_1}{dt} = \frac{dx_2}{dt} = \sqrt{2gh}\ \sin\alpha,$$

$$\frac{dy_1}{dt} = \frac{dy_2}{dt} = \sqrt{2gh}\ \sin\beta,$$

$$\frac{dz_1}{dt} = \frac{dz_2}{dt} = \sqrt{2gh}\ \sin\gamma$$

for corner drop.

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ANALYTICAL CONSIDERATIONS IN THE DESIGN OF A POLYURETHANE FOAM SHOCK MOUNT FOR POLARIS*

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This paper discusses pertinent aspects of designing a polyurethane foam shock mount and presents a mathematical model to simulate the mount's dynamic performance.

INTRODUCTION

The application of a flexible, foam material to the isolation of an item as massive as the POLARIS Missile required design approaches not covered by current literature on packaging. Most such literature delves into efficiency or cushion-factor theory, or material classification according to the maximum acceleration obtained under drop test conditions. Since the shipboard shock environment specified for design is quite different than a drop test, and since maximum acceleration is not entirely indicative of satisfactory shock mount performance, more specialized tests and analytical tools had to be developed for packaging the POLARIS.

One of the primary design tools, which had to be developed, was a mathematical representation to simulate foam dynamic behavior. This mathematical model was used to obtain system response via complex computer programing and, therefore, had to describe the non-linearity and rate sensitivity inherent in the foam. This paper shows how the foam characteristics were incorporated in the model and how these properties affected the shock mount design.

MECHANICS OF FLEXIBLE FOAM

An understanding of the mechanics contributing to the stress-strain behavior of foam is an important factor in designing with this relatively new material. The type of foam discussed in this paper may be described as an open cell, low density (4 lb/cu ft), flexible polyurethane foam. Its microstructure is a geometric arrangement of inter-connected ligaments with thin membranes stretched across the ligaments to form cells. In such an open cell foam most of the membranes are ruptured allowing flow of air through the structure.

The typical non-linear, static, compression-deflection characteristics of this foam (Fig. 1) are the result of elastic buckling of the ligament structure. The initial compressive modulus is related to the elastic deformation of the structure before buckling, and the plateau is where massive buckling of the cell structure is in effect. The bottoming phase occurs when the ligaments finally pack down on top of each other.

Shear and tensile properties of foam (also shown in Fig. 1) are essentially linear, indicating that massive buckling of the structure is not a dominating phenomenon in these modes of deformation. The shear and tensile properties are of interest in cases where a tensile bond is maintained at the foam boundaries, but as a rule, the nonlinear compression properties are of main concern in most packaging applications. Therefore, only compression properties will be discussed henceforth.

In the design of a shock mount, the dynamic or high strain rate properties of the material are of the utmost importance. Rate sensitivity of a foamed elastomer is very pronounced and is the resultant of two effects. One is the viscoelastic rate sensitivity of the parent elastomer, and the other is the pseudo-viscoelastic effect of air compression within the cell structure.

*This paper was not presented at the Symposium.

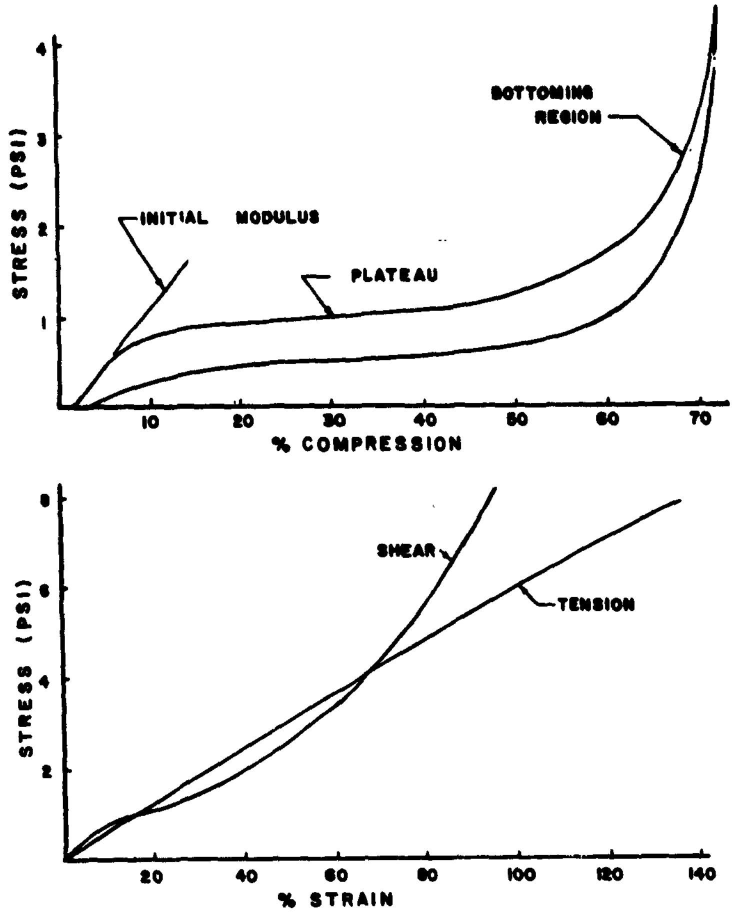

Fig. 1 - Typical stress-strain properties of polyurethane foam

Under impact or high strain rates the parent elastomer becomes stiffer (as is the case for most such viscoelastic substances), and a higher plateau or buckling stress is produced. At the same time, air in the foam is compressed and forced out through the many passageways in the structure. Due to the orificing effect of the cells a pressure buildup occurs in the foam and a relatively high dynamic force is generated. This pseudoviscoelastic effect due to air compression is largely dependent on size and configuration of the foam package and can be the dominating effect at strains over 20 percent, under dynamic or high strain rate conditions.

It is helpful to know the relative contributions of the elastomer viscoelasticity and the air compression effect under dynamic conditions. One method for determining this is by the use of time-temperature superposition. It has been shown that data analysis techniques described in work by Williams, Landel, and Ferry[1] can be used to interrelate stress-strain measurements as functions of time (rate) and temperature. In other words, with the proper time-temperature correlation, an equivalent high rate stress-strain curve (with air effect excluded) can be obtained by making a static or low rate test at a given low temperature. Figure 2 shows the "static" stress-strain characteristics of a foam at 0°F. According to the time-temperature correlation, this temperature is equivalent to a strain rate of 12 fps. A typical dynamic curve obtained from drop test data is included in the figure. The difference between these two curves is the air compression effect.

[1] Williams, M. L., Landel, R. F., and Ferry, J. D., J. Am. Chem. Soc. 77, 3701 (1955).

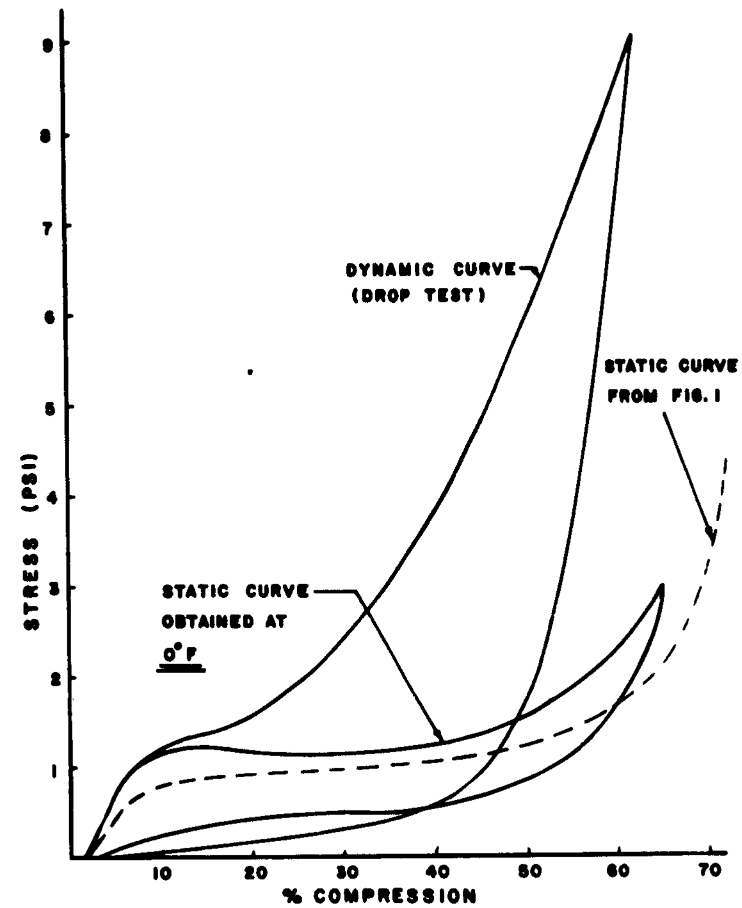

Fig. 2 - Dynamic properties of foam

SHOCK MOUNT DESIGN

In the POLARIS system, an annular space surrounding the missile was provided for the lateral shock mount. Design of a foam mount for this configuration was found to be influenced primarily by the air compression effect as discussed in the preceeding section. This was due to the large area of the packaging surface and the severity of the specified shock velocities.

An important tool in the development of the foam mount was a prototype-sized shock test apparatus which provided realistic design information and verification of basic predictions and assumptions. This apparatus, called the Partial Full Scale Test Machine (PFSTM), is a drop-test machine scaled to represent a vertical section of the missile and shock system. The essential features of the machine are shown in Fig. 3.

Shock mount performance is evaluated according to the dynamic force-displacement characteristics obtained from this test. Typical responses are shown in Fig. 4. Curve A is the response for a continuous foam package around the diameter of the missile. This is referred to as an unventilated configuration. Curve B shows the effect of cutting slots or air chambers around the diameter to promote free flow of air out of the foam. This configuration is described as ventilated. These comparisons illustrate the importance of configuration and the air compression effect.

In order to control the air compression effect more closely, and to increase the static stiffness of the mount, a three-segment air bag configuration was developed. In this configuration, the foam is split into three 120-degree segments per tier (Fig. 5) and covered with air-tight bags. Displacement of the missile

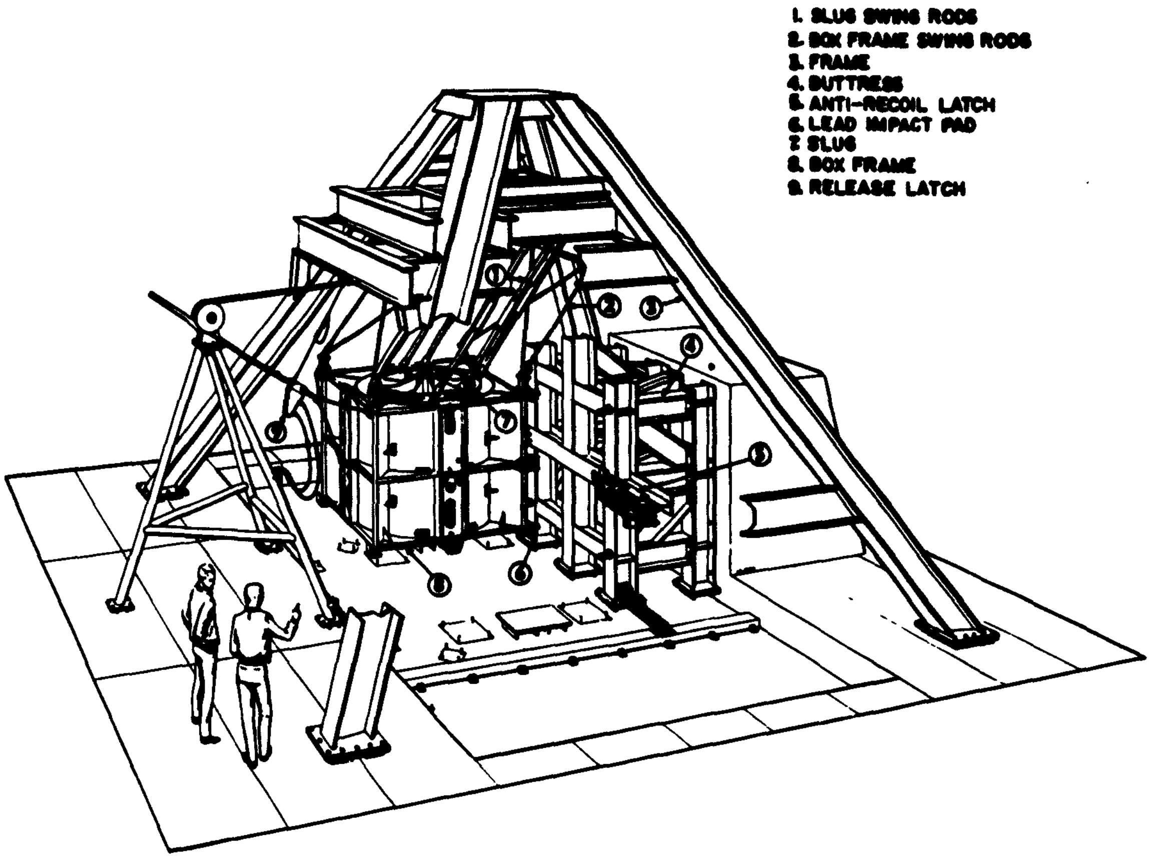

Fig. 3 - Partial full scale test machine (PFSTM)

builds up air pressure in the bags such that the static characteristics of the system are stiffened and more closely approach the dynamic characteristics. Also the air-bag configuration tends to regulate the air compression effect in the foam such that small changes in ventilation due to manufacturing tolerances will not affect dynamic response.

The static and dynamic force-displacement characteristics obtained with the air-bag configuration, foam, shock mount are shown in Fig. 6. The stiffness of the shock mount for small displacements, and hence, the vibrational characteristics, correspond to the initial compressive modulus of the foam. For small displacements, the volume change is not enough to build up any appreciable air pressure in the bags, and all the spring force must come from the foam structure. For larger displacements the air pressure buildup in the bags governs the shock mount characteristics.

MATHEMATICAL REPRESENTATION

An adequate mathematical representation of the shock mount just described must take into account the mount's nonlinearity and rate sensitivity. It was felt that the three parameter mathematical model (Fig. 7) fulfilled these requirements. The model consists of a nonlinear, static spring, k_s, and a Maxwell element denoted by k_1 and c. The characteristics of spring, k_s, are defined in the figure and were obtained from static tests on the PFSTM. The physical effects of the initial compressive modulus and buckling of the foam structure, and the pressure buildup in the air-bags are described by this non-linear spring.

The viscoelastic behavior of the elastomer as well as the pseudo-viscoelastic effect of air compression are taken into account by the Maxwell element. The Maxwell element adds to the static spring, a stiffening effect which is a

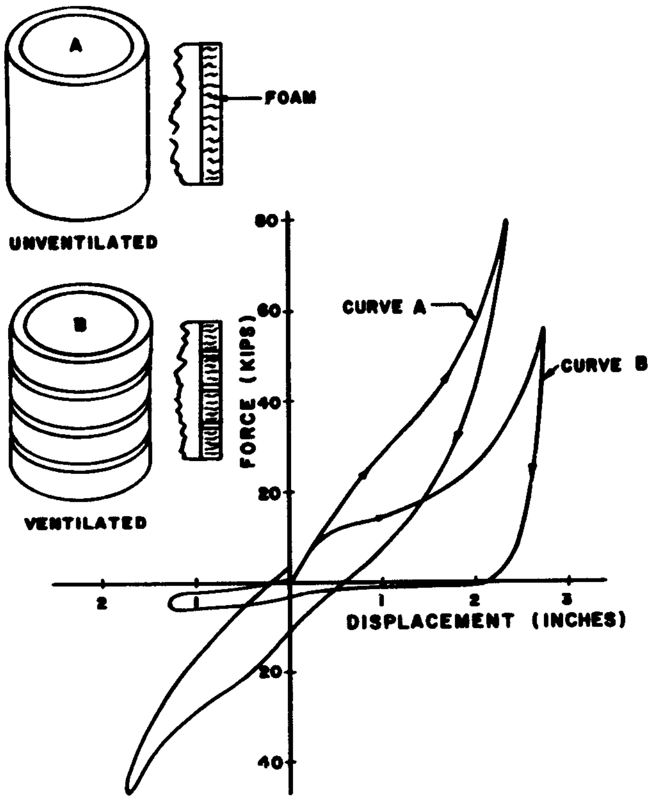

Fig. 4 - Effect of ventilation on PFSTM dynamic test results

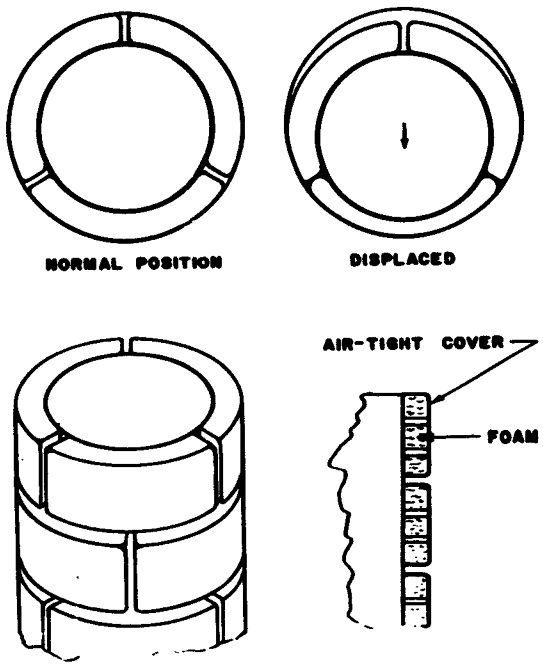

Fig. 5 - The three-segment air bag configuration

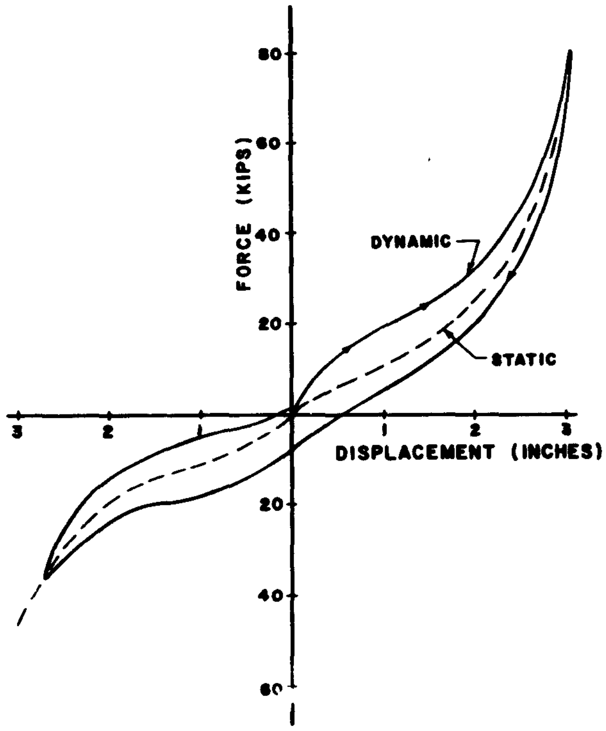

Fig. 6 - Force-displacement characteristics obtained with the air bag configuration

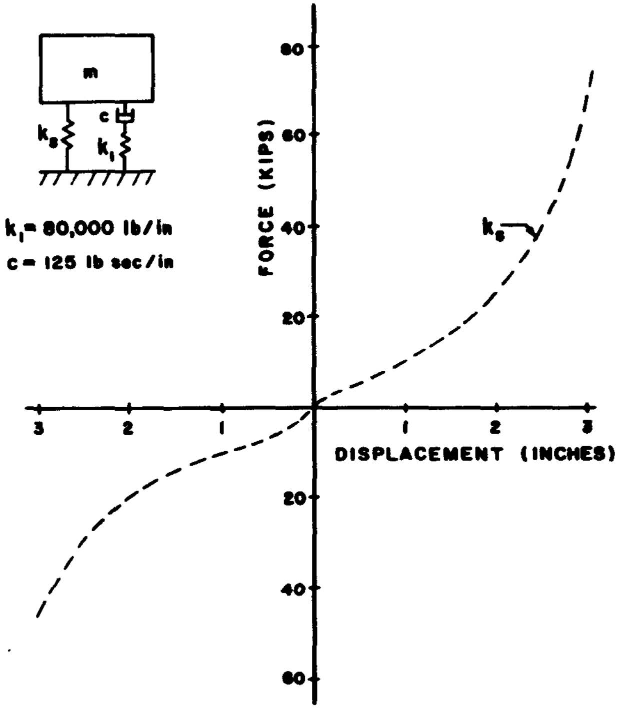

Fig. 7 - Mathematical representation of the foam shock mount

function of velocity. This element also provides the damping to simulate that which is inherent in the foam mount. The values used for k_1 and c in the Maxwell element are obtained by curve fitting dynamic test data from the PFSTM.

The calculated response to a 6-pfs step velocity obtained with the model given in Fig. 7, is shown in Fig. 8. Results obtained from the model appear to match the test data shown in Fig. 6 within the limits of test accuracy. Therefore, it is felt that application of the model to system studies is valid and that the shock mount performance under more complex shock inputs may be calculated with confidence.

CONCLUSIONS

A direct analytical approach to foam, shock-mount design does not appear to be available at this time. Because of factors of size, configuration, nonlinearity of foam properties, and so on, a semiemperical method is required. In this method, an understanding of the mechanisms of foam behavior backed up by a meaningful test program provides a rational basis for design. Buckling phenomena, pneumatic effects, and elastomeric rate sensitivity constitute the primary considerations.

A significant tool used in the design of the POLARIS foam shock mount was a mathematical representation of the mount's dynamic behavior. A mathematical model consisting of a non-linear, static spring in parallel with a Maxwell element appears to provide adequate representation. It is felt that this model incorporates the essential physical phenomena apparent in the shock mount.

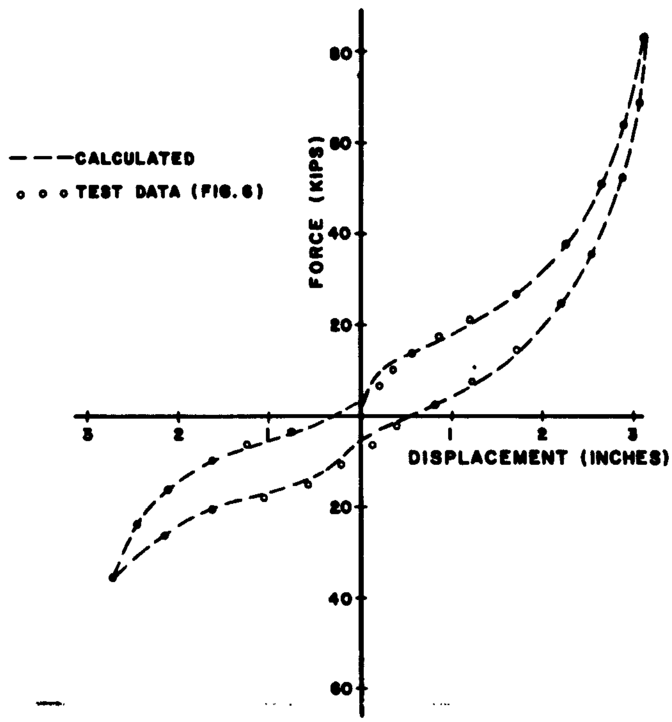

Fig. 8 - Calculated response obtained with the mathematical model

* * *

Section 5
PANEL SESSION II

THE OPTIMUM BALANCE BETWEEN COMPONENT AND SYSTEMS TESTING

Moderator - J. H. Armstrong, U.S. Naval Ordnance Laboratory, White Oak, Silver Spring, Maryland

Panelists - J. A. Harvey, U.S. Navy Underwater Sound Laboratory
E. Lunney, AVCO Corporation
S. Silverberg, Aerospace Corporation
A. Steinberg, Marshall Space Flight Center, NASA
R. Switz, Space Technology Laboratories

At the beginning of the session each panelist made a short opening statement to indicate his interests and his position on the panel topic. An edited version of these opening statements is included. The remaining discussion has been summarized and rearranged under appropriate subject headings. All comments were edited for clarity; a few comments were not included, either because they were not recorded or because they repeated a point brought out earlier in the discussion.

OPENING REMARKS BY PANELISTS

Mr. R. Switz (Space Technology Laboratories):

Since I am not an authority on other environmental problems, my remarks will be limited to spacecraft and missiles. Furthermore, I'm going to try to appeal to the people who write specifications and develop test philosophy rather than to those who actually conduct the tests, since the arguments that I will try to develop can better be used before the specifications are written. Finally, I expect to limit my discussion to qualification testing or that testing which is contractually required. Engineering development tests are not included.

Those of you who have worked with the Air Force know that, in many cases, a specification states quite clearly what is to be done even though it doesn't specify how it is to be done. The kind of specification I'm talking about, for example, would require that a battery, having a certain part number, be subjected to a series of specific tests set forth in the document.

I favor the component test philosophy over the systems test philosophy. With respect to a systems test, both the electronics people, who are quite interested in achieving their goals, and the structures people, who furnish the test bed for the electronic components, are concerned. All too often our experience has shown that there is no one test that will satisfy the objectives of both groups. In spite of this, usually only one test is performed, therefore, both positions are compromised and neither party is satisfied with the results.

Within the present state of the art, nobody really knows the input to the re-entry vehicle or satellite, in a spacecraft. It has not been measured and it is doubtful that it will be measured, since, by and large, the instrumentation flown consists of such things as piezoelectric accelerometers. A transducer of this type measures response, not input. It is wrong to extrapolate these responses and infer that they represent the input to the entire system.

Even if the input were known, it would be questionable whether the loads could be reproduced on ordinary test equipment such as electrodynamic shakers. The most severe loads are encountered during liftoff and during Mach 1, or max q. These severe environments are not caused by engine pulsations, but rather by aerodynamic turbulence, buffeting, acoustics, and so on, which cannot be simulated on an electrodynamic shaker.

My last point in support of component testing is the fact that our systems are getting larger all the time, thus some means other than a Go, No-Go systems test must be used to establish reliability. There are uses for systems tests in other than a qualification role, for instance, in quality control checks. In these cases, the systems test would be used to check workmanship and life, thus not requiring the input to resemble the environment during flight.

That summarizes my position and the points I would like to try to defend.

Mr. E. Lunney (AVCO Corp):

My background is associated with the design and development of re-entry vehicles for an organization working with Air Force contracting requirements and specifications.

The process of developing the components and systems of a re-entry vehicle involves three testing phases that follow a logical pattern. The first is called design data testing or, more commonly, development testing. In this phase it is primarily the designer who dictates the levels of the environment, the number of items that he has available for test, and the depth to which the development testing goes.

Development testing is primarily concerned with the selection of either vendor's, or in-house designed and fabricated, components. The idea is to try to scout out the failure modes and, you might say, the margins of safety in the performance area. Systems are not usually tested at the same levels. The development test levels may be either equivalent to or lower or higher than those used in qualification tests. In any case, development testing produces the best candidates for a reference design.

The reference design components are then ordered in some quantities and we proceed to the next phase in which tests begin to become formal. This phase is called flight proof testing and its purpose is to demonstrate that these components, integrated as systems, are flight worthy as far as the first flight tests of the re-entry vehicle are concerned. In the case of some one-shot items, we may subject only 25 to 50 samples to the critical environments, such as shock, vibration, temperature, and humidity. Although this gives one some feel for the confidence associated with the particular design prior to flight, it is not complete qualification testing. We do, however, issue a formal certificate once items have passed this level of design maturity.

The next phase is the formal qualification testing which is primarily concerned with operational components. Here we test for probably 15 different environmental conditions with sample sizes ranging from 5 to 10 for components and 1 to 5 for systems. This is a contract required test. We issue qualification certificates for the items once they have successfully completed the qualification test program.

It is necessary to realize that, neglecting ground environments and storage conditions, the mission cycle of a re-entry vehicle is very short—less than 1 hour. Our systems, therefore, have somewhat different life problems than some of the other systems associated with naval or aircraft operations.

Mr. J. Harvey (USN Underwater Sound Lab.):

My background is related to shipboard propulsion and operating equipment, primarily with respect to acceptance tests for the Navy. But an important phase of our work is devoted entirely to development items that do exist and, regretably, are operational in the Fleet. This is gear which has been accepted and for which the manufacturer is no longer liable, so we're not bothered with the problems of contractual responsibility. It is the type of item that the ship needs to go to sea, and the item is built of components which, individually, have all passed general qualification tests to a general specification. Yet, as a part of a system operating aboard a ship at sea, the item doesn't work.

The type of testing we do is much in the manner of a coroner trying to reconstruct the crime. We take this existing system, bring it back to the lab and try to find out why it doesn't work at sea. Although the components have been tested to a general qualification test, there is no indication that the inputs are related in any way to the inputs the system gives to the components. For that reason I'm against this type of component testing. In general, it is impossible to use component tests for our type of trouble shooting. There are specific instances for which component testing is a very necessary tool, but not in this relation.

I think that one could say it's axiomatic that if the system must be reliable, the component must be reliable. I don't know whether or not this stems from probability theory, since I'm not a statistician. I try to tell the truth. It is well to remember, however, that a system of 500 components, each with 99 percent reliability, can be shown, on paper, to have a system reliability of less than 1 percent. In my experience, I have yet to see a system made up of qualified

components that has worked satisfactorily unless the system had been tested as an entity. The synergistic reactions, the effects of combined environments, will almost invariably cause some sort of failure. For this reason I am steadfastly against component testing alone. Any test which produces the same damage as the service environment is justified.

Mr. S. Silverberg (Aerospace Corporation):

I would like to limit my remarks, as did Dick Switz, because my experience has been strictly in the satellite and re-entry vehicle field. I don't know whether these remarks will be applicable for shipboard equipment or systems; I feel they might, but we'll let the people who work in that field make that decision.

Of necessity, some components become available for testing before a complete system can be assembled. On this basis alone, even under tight schedules, some component vibration testing will and should be performed, particularly for new systems. If, because of limited hardware availability, all these components must be used in the operational system, the vibration level and duration must be restricted so as to protect the useful life span. Otherwise, at least one component should be tested to levels which simulate, as best possible, the component environment plus a reasonable factor of safety. This will increase the confidence in system reliability before any system testing is performed.

A system has dynamic characteristics which manifest themselves as resonances at certain frequencies. The equipment components in a system are, therefore, subject to a vibratory environment which is strongly dependent on the assembly characteristics. These characteristics and the details of the exciting force are generally not available during the component design. It is undesirable to penalize the majority of components because of high transmissibilities to a few. But the philosophy which seems to have the most merit is to overdesign the components and thus build reliability into the system.

In the final analysis, the purpose of component testing must be to improve the system reliability. Following this philosophy whenever possible, sample components should be tested to several times the system levels. However, it is the system tests which check the functional capability of all the components arranged in the system. Other problem areas which may manifest themselves during a system test are structural integrity, loose wiring or interconnects between components, and disadvantageous arrangement of components. In spite of all the shortcomings of system testing, of which we are all aware, such as the inability to predict and apply the exact environment as Dick Switz mentioned, and regardless of what level of testing of components has been allowed by the schedules and financial situation of the program involved, it must be the system testing which indicates the reliability of the system and the prospects for a successful end use.

In summary, it is my belief that before end-use, the test which must be performed is the system test. The minimum level of this test to prove successful system end-use, must be the anticipated use level. Therefore, the system must be built to survive the anticipated environment twice. Testing of the components, which, as I have said, is done for the sake of the system, must rank second in relative importance.

Mr. A. Steinberg (Marshall Space Flight Center):

Having been in this reliability field for some time and having worked with such people as Dr. Lusser, I think you have some idea of the things I may be saying. In essence, I'd like to bring up some of the factors concerning reliability policy, both with the Army Missile Command at Huntsville and the Marshall Space Flight Center. Since I've worked for both groups, I recognize the differences between the two and would like to relate these. The differences aren't too great, actually, since the policies emanated out of the same reliability group. If you recall, a few years ago this was one agency at Huntsville, not two different organizations.

We have quite a few reliability policy statements, but the ones that are pertinent to shock and vibration are first, that we require in all our development projects an early estimation of environments; this should be corrected as soon as the systems can be instrumented and the operational environments measured. Second, the contractor, or the developing agencies if it's an in-house project, should work up a consolidated test plan to include both component and systems testing. Our contracts in Huntsville have reliability requirements that are related to these policies and which dictate that components be developed first and that a safety margin be demonstrated to be inherent within these components. Then, of course, the system can be fabricated, based upon the use of these high reliability components. We end up testing the system itself to show that we still have this desired reliability.

Because of limitations on time and cost, a component test program is never as comprehensive as

we would like to have it. The program variations have to be based on the nature of the system and the mission. For this reasion, I would like to categorize components in terms of how they should be tested. First, we have a standard type of time oriented equipment which has to perform month after month or year after year. This equipment can be specified in terms of mean time between failures. Next would be the one-shot type of equipment, such as the solid propellant rock or the thermal battery, where, in order to find out whether it has a safety margin, you destroy it in the performance test. Finally, there is the third type which has only one opportunity to perform in a mission, but can be tested over and over again on the ground. This type is exemplified by the liquid propellant engine, where dozens of static firings have been run before that particular engine is finally fired to loft the vehicle.

The MAULER and SHILLELAGH missile systems are two excellent examples of how policies are being implemented, particularly for the Army side of the picture. Both of these systems have test-to-failure programs for component development, which seem to be working out pretty well. The systems contain both electronic subsystems that are time oriented, and one-shot devices such as solid propellants. The two programs began with the usual procedure of developing a test plan, estimating the environments, and selecting critical components for test-to-failure under critical environments.

Critical components are those which can cause system failure by reason of their functions within the system. The critical components to be selected for the test-to-failure program should be either new or nonstandard, have an unknown environmental response, have a history of failure, or be a high population item within the system. Critical environments generally are shock, vibration, and temperature. In special situations involving such things as uncaged semiconductors or solid propellants of the double-base grain type, humidity can also produce serious deterioration.

The Mauler test program began with components because decisions had to be made concerning their selection and because, at the beginning, the system did not exist. Less than 2 percent of the components were subjected to a test-to-failure program, but in one case the Army was saved $5,000,000 because this program uncovered a marginal capacitor.

A component test program does not eliminate the need for environmental testing of systems or subsystems. As soon as they are available then, for all practical purposes, one abandons the component tests and goes to the next level of testing. I, however, would like to mention one thing further regarding component versus system tests. SHILLELAGH began with the same sound planning involving the test-to-failure as a basic contract requirement. Because of delays in test schedules and for other reasons, the contractor was forced into system tests long before he was ready, and at a rather fantastic rate of expenditure, compared to the original cost estimates. These early system tests were very unsuccessful. The program was halted and then reverted to an applied research effort involving component test-to-failure. This effort brought component reliability up to the required level and resulted in a highly successful systems test program. Early component test-to-failure does pay off.

At Marshall, we are concerned with large, liquid-fuel engine systems and there are several restrictive problems. First, the cost of hardware for a large liquid-fuel engine is almost prohibitive. Second, there aren't many such engines in existence. Finally, the worst problem of all seems to be the difficulty of predicting and simulating the component environments involved with these engines.

There are three phases of testing involved: the PFRT, or preflight certification test; the QUAL, or qualification test; and finally the acceptance test, which is a test that has to be run on an engine prior to delivery or acceptance by the government. These, of course, require demonstration of performance to the safely limits as well as to environments. The component environmental tests are generally conducted after PFRT, rather than before, so, in this case, we go into system tests first and then to component tests.

The major sources of vibration in the liquid-fuel engine are either from the rotating motion of the turbo pump or from the combustion process in the combustion chamber. Also, since we gimbal the engine for guidance control, some translatory motion is produced. In any case, because of the mounting arrangements of the pump and the variations of the propellant burning, it is almost impossible to predict the vibration spectrum and magnitude. For example, we recently had a design review meeting on one motor at which it was announched that 1000 g's at a frequency of 3600 cycles had been measured. This had caused some breakage in the regenerative tubes that make up the thrust chamber and also, there was some leakage. The cause was attributed to a structural weakness. However, this same engine had been subjected

to shake-table tests of the magnitude decreed and nothing had occurred to show that this weakness could crop up during the combustion process. This example indicates that there is something lacking in our ability to predict the vibration environment.

DEFINING 'COMPONENTS' AND 'SYSTEMS'

After the Panel's opening statements, Mr. Bentley of Lockheed suggested that, even with the panel members, what was a component to one was a system to another. He asked that each panelist give his own definition. The Chairman agreed and asked each panelist to say what was his largest system and what did he consider a component.

Mr. Silverberg replied, "When I use the word system, I have in mind an entire satellite vehicle system. This doesn't include the boosters or the Agena, but it does include everything on top. You can picture what the components are, then, on this basis. They are the guidance packages, not the individual things."

Mr. Harvey preferred to stay with the definition in Websters and say that a system was an assemblage of items all having interdependence or interaction with each other. It was difficult to draw the line, but, in general, he would call a relay a component even though there was internal reaction, and an assembly of parts on a common base, cabinet, or structure, he would call a system.

Mr. Switz agreed with Mr. Silverberg. He, too, thought of a satellite as a system, and suggested a thermal battery as a component. There were, however, many items in a twilight zone, for instance the attitude control system on a satellite. He believed that, as far as his interests went, about 80 percent of his quarrels could be clearly defined as either system or component quarrels. He did not think it would pay to spend a lot of time arguing about the definition of the remaining 20 percent.

Mr. Lunney said, "From my standpoint a re-entry vehicle is a system, but certainly a major assembly type of system. This could be broken down into an attitude control system and several other types of systems. An attitude control system is one of several subsystems which make up a complete re-entry vehicle. The components of a subsystem, such as an attitude control system, could be nozzles, gas storage bottles, and possibly some electrical controls. Going down lower in the assembly echelon, we have piece parts which I would call items such as condensers, resistors, chokes, and transformers."

Mr. Steinberg pointed out that the Army use of the term component was synonymous with piece parts. The lowest level item that could be ~~used as a~~ replacement part was a component. As an example, when last he heard, the Army had abandoned any effort to maintain gyros in the field, therefore a rate gyro would be a component. A resistor was also a component unless encapsulated, or entwined with a printed circuit when the circuit itself would be discarded by the maintenance activity. The application of the term component or part, therefore, was flexible, since decisions were being made every day on the levels of maintenance to be conducted on each system. He considered any combination of components which could be tested as a system, a subsystem or system.

Mr. Weixler, McDonnell Aircraft, suggested that it might come as a surprise to many that MIL STD 127 defined parts, modules, and so on. A relay for instance was a part; an armature was an element, and the definitions included subsystems and systems.

DEFINING THE TEST

Dr. Cook of Collins Radio suggested that, since the panel members were in favor of component or systems tests, they should say what tests they proposed.

In reply, Mr. Silverberg said that no matter what system was in mind, the hope was that one could anticipate the vibration that the system and components would experience by analyzing or obtaining data from similar, operating systems. It had been his group's experience with satellite and re-entry vehicles that the vibration of both systems and components had a rather wide spectrum, but with very sharp spikes comparable to discrete sinusoids. They made the best use they could of flight data—taken on hard structures, not on platforms—and they analyzed various conditions to get their system inputs. It was much more difficult to get component inputs since these depended upon the location, and the mounting bracketry used to attach the component. A rule of thumb was to use 5 or 6 times the system level plus a factor of safety. After that, the system test was done to find if the transmissibility was so high that perhaps the component test was not adequate.

Mr. Harvey said that first of all, agreement must be reached on the meaning of the word

test. If they were concerned with evaluation, then the aim was to determine if the item or system met the design requirements under its true service environment. That was what he called a test. If, however, the aim was a diagnostic examination, then this was more in the nature of an experiment. It was still called a test but it was not, in his opinion, a test in the true sense of the word. If it was an evaluation-type test, then he believed that it was the engineer's responsibility to duplicate the service environment by any means possible. If it was a development tool for the designer, then any means were justified to the end.

Mr. Switz interpreted Dr. Cook's question to mean what method can be used to arrive at tests to be used once it has been decided to test systems, or components, or both. As had been pointed out when flight data was available, this was fine, but the people designing APOLLO, for instance, did not have much flight data. "There have been rather successful attempts to predict, by using models and subscale tests, what the response is going to be at the component level for future systems. I think TITAN II is an example of this. Aerojet, Martin, and Bolt Beranek and Newman started some sort of a scale model program about 2 years before the first TITAN II flew. They predicted the in-silo acoustical environment and, using TITAN I experience, came up with some sort of a transfer function from acoustic noise to vibration. As a result of this, vibration specifications were devised quite early in the game for the TITAN II components. Now, as the static firing and flight data are starting to come in on TITAN II, we find that they really weren't too far off. So I think we are making some progress in this field. It isn't all black magic anymore. However, I personally still feel that the state of the art demands that we stick to predicting component responses rather than system responses."

Mr. Steinberg said that on the contract side of these development programs they had run into problems through trying to specify everything that would happen to the system before it existed. They had started by asking for every sort of environment and the result had been to place such a test burden on the development agency as to almost price it out of business. One test plan received from a contractor had had some 27 different tests planned for different modes of failure. This was impractical and the approach now is to consider only the critical environments. As for component test levels, it had been found impractical to specify either magnitudes or spectra; rather, it was more practical to allow the contractor to develop his own specifications, predicting the most critical situations by his own theoretical studies. Then, after completing the development phase, the government would go through the normal qualification and acceptance testing.

Dr. Cook thought that one of the reasons it was so hard to define a test was that everyone was too preoccupied with reproducibility. "We generally find ourselves in the position of having to qualify something to a test which has been arrived at through a great deal of hocus pocus and at levels to which people can test and hope they can claim reproducibility, so that someone can perform the test in another place and verify these results." He suggested getting away from trying to set reproducible levels and taking, instead, further steps in randomization, not only random vibration but the random application of humidity and temperature, at random levels. He thought this could be done within the environmental levels of use. Reproducibility was only something connected with the test, whereas the aim was to get something that would prove the system in its use environment.

To which Mr. Lunney replied, "One of the problems associated with randomization or combination testing is how, when you get failures, do you establish what was the primary mode of failure? Was it due to the vibration, or to the shock, or can you positively show that it was due to the combination? This usually precipitates an investigation of the serial or sequential type of environmental testing to find out what was your primary failure mode so that appropriate corrective action may be taken. At the system level, I feel that combined environmental testing is fine, but I think we've got a long way to go before we have enough, let's say, testing maturity to fully understand its implications and really get the most out of it.

MEETING THE TEST REQUIREMENTS

Mr. Jones, Admiral Corporation, spoke up for the manufacturers of 'little black boxes' or components. He had the impression that the proponents of system tests, while they were not sure what their system would experience, nevertheless wanted to test these systems to some environment which they had dreamed up in the hope that it would simulate the actual environment. Those in favor of component testing seemed to advocate overtesting to see just what would tear them apart. From the point of view of the person who had to build the components and somehow to supply a guarantee with them, he had a question. What happened if, after having been built to a particular test specification

and having been put in a system, the component then fell apart?

The Chairman suggested that the manufacturer would still get paid, but would also be given a hard time.

Mr. Steinberg said that in all their reliability or test specifications there was no penalty for not being able to meet the spec except that the test had to be repeated. "As Mr. Armstrong mentioned, you still get paid. The only hardship on the company and the contractor is that they repeat the test without fee. So I guess this hurts the stockholder, but it doesn't necessarily hurt the employee."

Mr. Silverberg replied to Mr. Jones. "It's not as much black magic as it may seem. Even for new systems there are tools available to us which allow us to come pretty close to predicting what environment these black boxes will actually experience. We have wind tunnel tests, ground firings, previous flight data, and analytical methods. Usually you will find that the people who are writing these specs have had a little experience in the field of vibration, including experience with respect to what has happened with previous specifications that they have written. Many vehicles that the Air Force is flying now, or will be flying in the near future, use the same booster systems; therefore, as you go from vehicle to vehicle the things that affect you are things like external configuration, weight, and structural stiffness. However, you can get a good indication of what the components and what the system will experience. A good point was brought out that specifications written on black boxes are written to a high enough level so that when the black box, or component, is put into the system, we normally anticipate that it will not fail if it has passed its own qualification test. Now that doesn't say it can't fail, but this is usually why the black box test is as severe as it is. At some place along the line, the subcontractors have to be able to sell their equipment, and it would be ridiculous to make the test level so low that we could anticipate trouble just because of normal transmissibilities in the system."

Mr. Switz commented that some problems could be traced to component manufacturers who failed to put the specification numbers into their design.

Mr. Harvey said he had the feeling that the space people wrote specific specs for most of their black boxes because they could get a pretty good idea of what the system input to the black box was going to be. This was not true of shipboard gear in general, the majority of which was built to very general specifications. In his opinion, the fault lay in the specifications themselves, which were too loose and general for application to specific places aboard ship. Ship gear was tested to extremes, usually, he believed, higher than necessary, and was then put into a system in which, because of internal coupling and interactions, had all hell shaken out of it so that it fell apart, but this shaking was quite different from the original test.

He wondered if it would be possible to give, for ships, specific information for each location, but he doubted this since there were so many systems. Perhaps better or more standard models for systems should be tried. There should be a feedback circuit between the general specification and the manufacturers of specific items.

Mr. Silverberg thought the space people were not as clever as Mr. Harvey thought. They did not write specifications for individual black boxes.

The Chairman suggested, "Maybe what's needed in some cases is a little footnote under each clause in the specification which says whether or not you really mean it. I'm reminded of the fellow standing right in front of a No Smoking sign as he lit a cigar. His friend said, 'Don't you see that No Smoking sign there?' He says, 'Sure, but it doesn't say positively.' "

Mr. Rouault, General Electric, wanted to pursue a point Mr. Harvey had made, the discrepancy between Navy specifications and the actual environment. He believed, from close acquaintance, that there was, unfortunately, little agreement particularly as to temperature, shock, and vibration. Many ships had been shocked, most of the equipment had survived, though some had failed, but what seemed most odd was that TV sets in the wardroom and small radios in the crew's quarters suffered no damage. Obviously these equipments were not built to Navy specs, but to good commercial practice. Again, he would like to ask why was it impossible for a man to stand on a vibration table or shock machine during a test without becoming a casualty?

Mr. Harvey agreed that there were discrepancies, but believed that something was being done to correct this situation. He went on to defend the Navy shock machine; "For many years I've heard remarks about the nastiness of the Navy medium weight hammer. I've heard people ask, 'Why don't we use a nice shaped pulse? Let's get repeatable results.' I'll go

along with this if we can get our potential enemies to use depth charges that produce shaped pulses." In the meantime the machines should be used. The machines were designed to produce damage similar to that experienced in the real environment. An excellent correlation existed between gear which had passed the medium weight shock tests and had later worked satisfactorily during shock hardening trials at sea. The proof of the pudding was in the eating.

Manufacturers were not given instructions on how to build something to be tested on the machine and stay in one piece. It was, therefore, important to get information back to the designers after a test. Mr. Harvey's division at the USL had recently been insisting that manufacturers use instrumentation during tests. While companies often insisted they did not care about instrumentation, they were back the moment anything broke, asking for the readings.

Mr. Rouault asked who was working to correct the specifications* and went on to relate a story about an equipment built during the war. After some 3000 of these items had been produced and had worked very well, a spec was written by the Air Force and then the equipment was tested. It disintegrated. In wartime the solution was easy. Two flight officers were brought in and two chairs were welded to the vibration table. The officers agreed that the equipment couldn't take it, nor could they, so the spec was altered. He was concerned that there was no such approach today.

Mr. Weixler said he had gotten the impression that only items that were overdesigned passed the tests today. Was this correct?

Mr. Silverberg replied that items which pass tests today are overdesigned, in parts of the spectrum. If one assumes that the test levels for components were justified at one or at many frequencies, he then finds that in the tests a sinusoid may be superimposed on the random excitation and swept across the spectrum. It would be true to say that this component would not experience this level at all frequencies in the field. In this respect then, only an overdesigned component could pass the test. "This costs us weight, time, and money at the component level, but it saves us at the system level. Remember that I think of systems in terms of satellite vehicles and in terms of literally millions of dollars. If you lose useful information from a system because of the failure of two components on it, you feel rather ridiculous. We try to avoid taking any suck risk. At the component level, we do overtest severely. At the system level, I mentioned that we test as closely as we can predict to the level of a flight, because some failures in the system just won't occur unless you get up that high. There we penalize ourselves, in that we design a system that has to experience the environment twice. This too, isn't a happy thought, particularly weightwise in a system. We really scrimp and just barely get that quality and reliability, which is our goal, into the design. I think that at the price you pay for the system, you are willing to overdesign the small parts."

Mr. Weixler replied "I wonder if you mean overdesign or designing with Lusser's testing concept in mind. Lusser says that you find out what the ultimate stress is and measure your safety margin below that. I believe that's what you mean when you say you design to twice the anticipated environmental stresses."

Mr. Silverberg: "I didn't say to design to twice the anticipated environment. For the components I design to about five times the system's environment, but I anticipate that at some of the frequencies of the spectrum the component will actually experience these environments. The difficulty is that, in the test, the component will experience these levels at all frequencies, while, in the environment, they will only experience them at particular frequencies. From the point of view that there are frequencies where you are designing to high levels that do not occur in flight, the thing is overdesigned."

The Chairman made the point that test procedures and the specs which called for them were not moral questions. It was not a question of fair or unfair tests, but of useful tests and tests which were not useful. The basic purpose was to achieve a satisfactory level of reliability. Anything which helped toward this end would be appreciated.

Mr. Armstrong went on to describe a management technique which might be of use in the controversy between designers and testers. "I am particularly interested in this technique because I am its victim. I spent 14 years in the shock and vibration test business, mostly concerned with rather high level shock. I don't really feel comfortable at less than about 5000 g. At the end of that time, when I had made a large collection of friends among the design groups and had been called many things, I was put in the design business where I've been for

*Improvements and amendments to Navy Shipboard S&V Specifications are under study by the Bureau of Ships, Code 423.

the last 5 years. In my case, it certainly has given me a great appreciation for both sides. Now I look back to see who is responsible for the horrible test which is beating up my designs and find out that I have nothing to say. Maybe this technique could be used in some isolated instances."

Mr. Davis, General Electric, raised the subject of acoustic testing. At General Electric, they wished to simulate the high level vibration experienced at launch and at max q, but they realized that the main cause of the vibration was the acoustic environment at launch and the fluctuating pressures induced by the turbulent boundary layer at max q. They were giving serious thought to the possibility of an acoustic noise test on a system as a better means of subjecting components to the vibration environment they were likely to experience in service. With an ordinary vibration test, the higher frequencies might never reach the components because of structural attenuation. With acoustic excitation, however, frequencies of 500 to 1000 cps may be seen by the components mounted on, say, panels directly excited by the noise. He asked Mr. Silverberg what he thought of such a procedure.

Mr. Silverberg replied, "If you could show me an acoustic chamber that could produce the environment you want instead of a chamber which gave only a certain number of discrete frequencies, as most acoustic chambers really do, and if you could show me a good test plan, I'm all in favor of it."

Mr. Switz brought up a point which he thought had been overlooked. "If you're on a system qualification basis, what happens when one of your components fails during your system test?" Should one replace the failed component? If one does so, is it then fair to subject the structure and other components to the same environment again? What happens if another component fails on the second test, and so on? How does one solve this problem? By his reasoning, there was no answer. The system route could be very costly both in time and money. He had known two cases where flight dates were missed, by months, because everything was 'hung' on the system test. The day for the test came; the system failed; and they had had to go back and redesign. Mr. Switz was convinced that it failed because they tested at levels which were too high.

ANALYSIS VERSUS TESTING

Mr. Nankey, General Electric, discussed the role of analysis. He said, "I feel a very important factor in this matter has been absent from the discussion; the role of analysis in the testing program. Analysis can be made at various design states—at the component level, at the subassembly level, at the assembly level, and at the entire system level. Distribution of analytical work done along the way determines, to a very great extent, how much testing is required at each level. We really have three ways to determine the progress of a design and to predict how well it will work out. We can use past experience, theoretical analysis, and testing. They are, to a certain extent, interchangeable. In many cases at the component level, we can make an analysis that completely obviates the need for test, or we can rely on a past experience. Past experience tells us that we will have vibration troubles if we install capacitors without any support other than their leads. Testing just isn't required in a case like this, since we have another means of determining the factors affecting reliability."

Mr. Harvey replied that he believed the next proposed issue of the Navy high impact shock spec was to make use of dynamic analysis, in some cases, in lieu of an actual test. He did not know when it would be out.

Mr. Lunney also commented on this subject. He said, "One of the motives for testing, especially qualification testing, that we've had drummed into us is that you are demonstrating that the performance of the system or component will meet detail or model specification requirements, not general specification requirements. I feel that dynamic analysis plays its major role in writing the specification to give you realistic design criteria and testing procedures. This is where analysis, especially in these way-out-yonder type of programs that we're in today, has to be employed, for example, in predicting the launch vibration environment for a command module on the moon, where we have an entirelv novel set of circumstances. This is where analysis can really support and defend, or generate, a testing procedure."

Mr. Bakalish, Martin Company, said he had interpreted Mr. Nankey's comments somewhat differently. At Denver, he was primarily concerned with failure analysis, or what he preferred to call the autopsy method, and he believed that this method was often ignored. His earlier concept had been that design engineers were ignoring the environment; now that he was amongst them, he found that both the environmental people and the design people were ignoring the obvious. "Just because we subject an item to vibration or shock and it fails, we say that that item has to be redesigned because of

the shock or vibration environment. The obvious point is that it was probably too poor a design even to operate on a bench without the dynamic environment. This is the point I think we are missing. We are putting the cart before the horse. We have to use this experience of failure and autopsy to determine, before we go into the environment, just what the proper design should be.

Mr. Nankey agreed. He wanted to question the panel on the relationship of analysis to testing, not to imply that any particular agency was at fault, but that not enough was being done. He felt this subject had a strong relationship to the proper balance between component and system testing. If an adequate analysis could be made, or if past experience with similar systems was applicable and the answer was known to begin with, testing need not be performed at every level of the design; however, when the final assembly of the complete system was reached, it would have to be tested. This would be the time to perform the qualification tests. He did not want to imply that this was not necessary.

THE STRUCTURAL PROBLEM IN THE SYSTEMS TEST

Mr. Davis said that the problem his group faced was that the systems test was a structural test. There had been cases where secondary and even primary structures in a satellite had failed during laboratory vibration tests. This could be very disturbing, particularly considering that the excitation in the test was only in one plane and not three planes as in flight. What was even more perplexing was that the same vehicle that failed in the laboratory had flown successfully several times with no indication of structural failure. Mr. Davis believed that, if one were to examine the loads put into the structure in the system tests and to try to extrapolate to what the loads would have been on the other side of the missile-satellite interface, it would be found that the missile should have broken apart well before the satellite did.

Mr. Switz thanked Mr. Davis for raising an important point. Mr. Switz felt that we got into situations such as had been described because all too often we took flight measurements and then inferred that these were representative of inputs to the entire system, rather than inputs to black boxes. These rather high levels were then used to generate long duration tests which people felt were necessary to demonstrate system reliability. Often the equipment people had not had enough experience with a certain part to know what it could take, whereas a man in structures could tell to four significant figures what a rivet could take. The point was that the structures people did not need the increased level of intensity and longer time which the electronic people needed. So his answer to Mr. Davis was first, that there was an unfair extrapolation from flight data; flight data should not be used in that fashion. Second, the boundary conditions in the lab, the shaker fixture, and so on, were probably very unrealistic and did not appear to the satellite as the booster did. Third, the load path was entirely different.

Mr. Switz reiterated his belief that the most severe dynamic loads were not imparted to a structure by engine pulsations transmitted the length of the missile, but were induced by higher frequency modes excited by aerodynamic turbulence, buffeting, and so on. He maintained that an electrodynamic shaker was not a simulation of this environment.

Mr. Silverberg pointed out that the reason vibration tests were specified in only one plane at a time was that, in general, systems were excited by a force at their base or at one point. To shake a system laterally as well would require input of a displacement and a rotation, and the ratio between the two would have to vary at all frequencies because the equation of motion of lateral vibration was a fourth order differential equation. Since this could not be simulated by putting in a displacement, they did not try at all. Nor could they, in the case of vibration, simulate aerodynamic forces distributed over the missile, however it was done for static loads where possible. Regarding the structures argument he had to disagree completely with Mr. Switz. He said, "I mentioned before that many vehicles use the same boosters and many of them use the Agena that the Lockheed Company produces. There is a great deal of flight data available on the Agena. For vehicles that use this booster, which have roughly the same configuration and weigh about the same as past vehicles, this data, taken on hard structure, indicates what motion was experienced at that interface. If I put that motion into the system, whether it's through a shaker or otherwise, and don't worry about the force needed to put in that motion, the motion above that station will follow appropriately at every frequency. The system isn't linear, whether it's being shaken in flight or not. I'm limiting myself to axial vibrations because I've already indicated that we don't do lateral vibrations."

Mr. Steinberg commented, "At the Army Missile Command we see that everything is strength oriented rather than time oriented. In our Army contracts we have a specification

requiring that the safety margin be demonstrated to four standard deviations above the reliability boundary. This means that, even for electronic equipment, we ask that a test-to-failure program be initiated to show that the safety margin exists. Of course, a safety margin initially means the risk is high and there is a gross lack of knowledge of the existing ambient environment in the mission. Where you start out with specifications and the spread is very great, the safety margins show a wider spread than safety factor. As you begin to produce the item, to maintain your quality control and begin to know something more about the ambient environment, there is less spread and this safety margin becomes less stringent than the safety factor. Either way I'm sure that there must be some compromise between everything being time oriented and everything being strength oriented. We require a contractor to demonstrate a reliability number. By the mean time between failures (MTBF) concept, a number is set up at a confidence limit to show that this MTBF has been attained. When trying to come up with techniques to demonstrate the safety margin and also a reliability figure, Mr. Jim Norman, former Technical Director at ARGMA, discovered a procedure based on the test to failure. The title of his report is 'Estimating Reliabilities of Function Stress-Strength Data.' Using our contract requirement of 4 sigma dispersion around the yield point will give a reliability of eight nines. I think this will be quite helpful and, of course, as our knowledge becomes greater and the people who use the product get more confidence and knowledge about the system, their risk becomes less."

SPECIFICATION REQUIREMENTS VERSUS FLIGHT TESTS

Mr. Stallard, AVCO Corporation, said he wished to expand upon some points made by Mr. Lunney and Mr. Bakalish concerning components. The component designer was usually presented with a contractual document which specifies, in detail, the environments the components must withstand. He then set about designing to these environments. There might be argument on how this should be done, for instance would one design differently for 20 g and 10 g? He, Mr. Stallard, doubted it and would just design to the best of his ability to withstand the military environments involved. Now because of the time scale, the component manufacturer would be building components and, in the meantime, the missile manufacturer would be assembling airframes and needing components to fill the holes. As a result, a number of components were often brought in on open rejection. These items would not pass acceptance tests, but somehow missiles were sold with these open rejection items covered by alternate design change notices. Then, after quite a few R&D vehicles had been fired without flight failures, it would become clear that the components in question did work. Mr. Bakalish had suggested that components not passing the test would not work anyway, but experience showed that some components did not pass the test but did not fail in flight. "Then, when it comes to the qualification of these components to the original documents in the contract, the specification levels are reduced. We are accused of making a contract or a specification to fit the component and I think this is done quite often. I think system tests in our large complex vehicles are more or less the flight tests themselves. It's just like in the airplane days when, after the taxiing up and down the runway, sooner or later you've got to fly. Nobody can argue too much against a successful flight. I'm going to ask if the panel has any comments to make on qualification testing of components versus actual flight data. If you do have a group of successful flights yet cannot pass component qualification, the component manufacturer will probably tell you that for a total sum of so many megabucks he will be very happy to redesign your component. I do not think any of the procuring agencies would want to cough up a few megabucks to redesign a component that flies in a missile successfully. If you are a missile manufacturer the re-entry vehicle is a component, so you can take your definitions of components as you wish."

Mr. Lunney replied, "In the re-entry vehicle design development effort that we're concerned with, the contractor prepares detail or model specifications which establish the design requirements for a component or a system as well as the test procedures to demonstrate that these requirements have been met. This detail and model specification is coordinated with, and approved by, the procuring agency through a negotiation to agree what the testing conditions are to be for the component or the complete re-entry vehicle. Therefore, you might say the contractor is just as culpable as the procuring agency whether the specification is a good one or bad one. In the present state of the art, a lot of these environmental conditions are pinned right down in the same way as some of the old aircraft environmental conditions. This being the case, we should have a feedback from flight testing to our specification people, to our planners, and so on, so that we are continuously improving specifications. This can be, and is being, done today on the detail and the model specification level for a particular weapons

system program. General specifications which are published by the military are ordinarily considered guidance documents, to be used really if nothing else is available. They are supposed to represent the current state of the art. Some of them have been updated, i.e., MIL STD 810 which still has a lot of loop holes, but at least is a step in the right direction. I do want to stress that I feel, honestly, that the contractor is just as culpable as the procuring agency when we start talking about good and bad specifications."

Mr. Sutphin, Martin Company, said, "We have been able to demonstrate in a flight test program that we can have 14 or 15 successes out of 15 flights. In that same flight test program, we may have had some of the components which were on open end rejection. The company who designed those components would be justly proud in saying that their components demonstrated flight capability in a flight test program, and that we should buy them regardless of whether or not they meet the specification. I feel that, although that may be true in some particular cases, it is very dangerous to assume that it is true in every case and the reason is quite simply this. In our flight test program, I was privileged to conduct a flight on a crisp cool morning in perfectly ideal terrain at one of our local test sites. Say, for example, that the missile had an electrical item which I know will malfunction at 225°F, now on my ideal day it may be perfectly true that that item will not see 225°F at my test site. If, however, I were forced to fire from an installation in North Africa, I might see 300°F on the same item. Now, I'm not willing to buy a component which will work ideally on my ideal day when I know that I will be required at some time to use that same item under adverse conditions." Mr. Sutphin said he believed there were two causes for this situation. One of them was the short time for design and fabrication under contractual obligation. The second lay in the difference between the general specification written for the weapon system and the specifications written for individual components and sent out to component vendors. The two were not easily compatible. The basic problem was the misinterpretation of input and response and the fact that these two were often inserted indiscriminately and without justification.

Mr. Steinberg wondered what was a successful mission. Recently he had been operating with flight vehicles where a great many malfunctions were tolerated and the system was still said to be successful. How, for instance, did one grade the static firing of a liquid engine? In one firing, a 1000-g vibration had been measured in the thrust chamber and, even though the tubes had exhibited structural weaknesses and there had been considerable leakage, the test was considered successful. This sort of thing could become quite a game. He remembered in the Army a few years back there had been a system considered highly unreliable. This problem was solved by increasing the miss distance by a factor of 3 and the system was suddenly reliable. Another example was a missile ground-based automatic checkout equipment, one of the most unreliable components of a system. Should this item be considered part of the system or not? It was a question of changing the ground rules. The initial specifications were written, then as experience was gained, opinions would be changed and some parts would be said not to be critical. Some parts might be deleted altogether and the flight would still be successful.

Mr. Harvey also spoke on the subject of accepting items under performance limitations. They were all under some form of pressure to get equipment built and in operation, but this did not help to get the real environment established. It could not always be fine weather when submarines went to sea. Often the boats went out and as soon as they returned the bits were taken off to try and find what had gone wrong. This troubleshooting was what he called true system testing. He did want to be able to simulate the damage experienced at sea and to be able to determine the actual cause of the malfunction. Surely nobody wanted to change that requirement.

Mr. Silverberg cited the case of a system that was having a great deal of difficulty in flying a successful mission. It had not yet finished its qualification tests. He thought it fair to say that the Air Force and Aerospace had learned a great deal from this experience and it would not happen again. Priorities had now been changed and reliability, which really meant testing to the hilt, had been placed in front. Air Force generals were now saying that this thing must fly right the first time. You could not do this with open rejection components. You did not take that risk.

Mr. Stern, General Electric, thought one basic consideration had not been mentioned. Who was to pay the piper? In the contracts he had been concerned with, the time from contract to delivery was very short and the costing had been done very carefully. There was no time nor money for give and take between the designer and test engineer. AGREE testing had been mentioned. His group was doing AGREE testing on a system and right now they had to

buy 16 vibrators. For another program, they would have to buy 20 more. Who wanted to pay for this? The answer largely determined what kind of a program one had, and what kind of testing one did.

Mr. Steinberg replied, "The government puts these requirements on you as a developing agency. The test programs are to fulfill these requirements and in many cases we fund separately for them. In the case of the large liquid-fuel rocket engines, we have a gigantic test program that is quite expensive. If you want a series of engines for a Saturn system and you would like to use these in flight, you have to give us at least 4-year lead time. In one of the new projects that we have underway right now, we are funding more heavily on facilities than we are on the development program. For this particular engine, the fuel alone for one static firing costs $300,000, so that the matter of economics is one of great concern. We try to program and plan for this at the inception of the program. As to whether it is more economical to do it on a component or a systems test basis, I have to go back to my Army experience. The system does not exist at the inception of the contract, so it is more economical to test at the part or component level at that point. Either way the burden is on the government to pay for all these things that we keep asking for. In most of the industrial concerns I've visited, I have noticed that on almost every shake table there is a little plaque saying, 'Property of the U. S. Navy - U. S. Air Force or U. S. Army.'"

Mr. Lunney commented, "I would like to go out on a limb and suggest that, as a tight schedule, minimum dollar cost program for qualification, you take a crack at systems testing. When you are dealing with a prototype, however, you are going to run into failures of components of that system. We've gone through this now in two programs where our systems have gone to pot because of components and interdependence failures. We then had to go back to component testing and corrective action after failure reporting, and so on, until we got the system qualified. As you develop design maturity and revise the design, your calculated risk should be continually decreasing because you are, hopefully, improving the product. So I think that system testing on prototypes is probably the most economical and also the short time way to, we'll say, qualification success, but you may fall flat on your face and end up doing a complete component program."

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Section 6

PANEL SESSION III

THE RELATIONSHIP OF SPECIFICATION REQUIREMENTS TO THE REAL ENVIRONMENT

Moderator - Dr. R. M. Mains, General Electric Company

Panelists - Mr. Paul Hahn, Martin Marietta Corporation
Mr. F. J. Lindner, USA Engineer R&D Laboratories
Dr. I. Vigness, USN Research Laboratory
Mr. K. Ruby, Motorola Inc.
Mr. F. P. Klein, Aerospace Corporation

An edited version of the discussion during this panel session follows. Not all comments were included since some repeated statements that had been made previously.

The remarks during this session were not such that they could be separated according to subject. The discussion is therefore setdown in the order in which it happened.

DISCUSSION

Opening the discussion, Mr. S. Baber, of Boeing, asked how combined environments should be handled where there are three planes of vibration and a programmed temperature and altitude cycle to simulate at the same time.

Mr. Klein responded that a specification usually does not call out this kind of a test when it covers acceptance, qualification, or flight proofing requirements. He indicated that the intent of the specification should be clear in the way that it is written. If a combined environmental test is required, it should be relatively easy to perform by conventional laboratory means. He asked Mr. Baber, "Doesn't the kind of requirement you are discussing relate to an R&D problem in your company?"

Mr. Baber said that actually he was concerned with deriving the specifications. "We're getting involved with running temperature tests up to 2000 or 3000 degrees, combined with vibration. We're running many tests on humans in which we must, prior to flight, simulate the complete profile. We must therefore come up with a specification to reveal the operational characteristics of the human or component."

Mr. Klein felt that Mr. Baber had brought up a very interesting point in that many of the discussions during the Symposium had concerned performing a test to provide the basis for a specification. In reality, the intent is that specifications should be based upon life profile which may start at the factory and run through the re-entry phase. Mr. Klein also thought that more should be done to define the environments by measurements, during this period, so that the specification can be written without regard to the ability of the environmental engineer or the test equipment to perform the test. Even though there may be limitations to these abilities, the requirement still exists and must somehow be met.

Mr. A. Lynn, of Lockheed, mentioned that he had recently run into a number of cases where tests required to prove a package had damaged the packaged article. He gave examples and said that, at Lockheed, they recognized that the parts would be damaged if they went through the cycle of package tests that were specified. Attempts to feed this information back to the services were unsuccessful. Mr. Lynn asked about coordination of specifications and whether the requirements in the specs were correlated with the problem.

Mr. Lindner responded that, in most cases, specifications are coordinated and that, in many cases, a general environmental specification is used as a basis for the particular item requirements. He asked, "Who specified the cushioning for these items?"

Mr. Lynn said that he didn't know who specified the package, but that it was not compatible with the basic specification for the part; sometimes this was a result of the item document being too highly classified. He complained that some designers will not accept responsibility for the full life of the item and, apparently, in these cases the packages are designed without any regard for the item.

Mr. Lynn also said, "The government itself is guilty of causing a great deal of difficulty because they have a requirement that, for any item over a certain weight, they issue the bill of lading. If an item is prepared for air transportation, the government bill of lading may, after the item is packaged, specify a freight move; this can cause great difficulties."

Mr. Lindner was not familiar with the bill of lading problem, but said that the main problem seemed to be one of communication between the design engineer and the packaging engineer.

Mr. Lynn agreed and said that organizational problems sometimes prevented communications until the final stages of a project. He again questioned the willingness of designers to accept responsibility for the item to the point of use and said that, if the interim environments were well established, there shouldn't be any difficulty.

Mr. Klein thought the answer to the question was even more fundamental. He said, "It is not the responsibility of the designer to design all of the tests. The intent of the military specification is to provide the applicable documents pertaining to the delivery of hardware. In addition, the customer is obligated by requirements in documents other than just the transportation document, such as the reliability document. The reliability engineer is responsible to design a life test requirement or to determine the mean time between failure (MTBF) of the equipment that is going in the package concerned.

"Apparently there has been an inadequate handling of the specifications in the contractual house. There are many kinds of tests, including R&D tests, to find certain weaknesses for the designer, and there is the reliability test. Without the reliability test you don't have a feeling for what can happen under repetitive conditions, as are described in the MIL spec."

Mr. Noble, of ASO Philadelphia, commented, "Many of the items that are designed for ideal environments are subjected to environments in transportation and handling that far exceed the operating conditions. It is for this reason that the packaging specs are written the way they are. By the way, the answer to the question of whether or not the packaging requirements are coordinated with design requirements is that they are. If they are not, it's an error."

Mr. Kerstetter, of NOL, wanted to hear some comments on early design criteria. He asked, "When we list environmental criteria as one of our design objectives, should they be only limits or should we specify exact tests?"

Dr. Vigness replied, "Normally, when you design something, you begin with your small components and would like to know what these will be exposed to. Generally, you don't have this information, but only information as to what the completed structure will be exposed to. You don't have specifications for the individual parts, so you make tests as you go along and see to it that these tests are always in excess of the specifications for the completed equipment.

"You are trying to get the equipment as rugged and strong as you can under practical, yet economical conditions. If it is relatively easy to make the equipment much stronger, you do so; if it is difficult, you go only to what you think the conditions might be.

"Sometimes you find that by shortening a wire or changing the natural frequency of something you can make it much better than it was before. It is these developmental tests that you work with. Run it through a vibration or a shock test, find out what fails, and correct it. Do that several times and be sure that it is at

least several times as strong as the values specified for the completed equipment. Don't worry if it's a lot stronger because sometimes this can be accomplished with only a little bit of effort. This just gives you a safety factor."

Mr. Rouault, of GE, suggested that the basic difficulty is that specifications are much too broadly written and live too long. He cited an example of a Signal Corps specification dated in 1886 for which waivers were still required. Mr. Rouault thought that, in order to overcome this difficulty, the problem should be defined, analyzed, and clarified by a reasonably broad technical study. This study should be the basis upon which specifications of limited life could be issued. Certain numbers, pertinent to the state of the art at that time, would be incorporated in the specification. If the specification were given a limited life, say 5 years, then there would be a means by which a change of the fundamental spec could be initiated by reference to the basic technical study. This, implied Mr. Rouault, would avoid the horrendous difficulty of specs which have long outlived their usefulness.

Mr. Hahn commented that the approach to design should be one of an iteration process where such a spec as MIL-STD-810 would be the first step in the process. He suggested the same type of thing to develop transient loads which end up as equipment environments. The environment finally used in a legal document would be the result of n load iterations. Referring to Mr. Rouault's suggested spec life of 5 years, Mr. Hahn said that, in some cases, the coordination process to get official acceptance takes longer than that.

He also said, "When you begin a program everybody is in a hurry. You subcontract things and, in order to get a subcontractor to build something, you must give him a legal document which includes a specification. He gets the spec and starts working. Then it begins to cost money and you want to tell him to redesign. The operating procedure should be such that you get better information as the iteration process continues, then, eventually, you know precisely what is required for each one of your individual packages."

Dr. Mains added, "MIL-STD-810 was written as a military standard so that it would not be a specification and so that it could more readily be kept up to date. They have very carefully avoided calling it a specification. Another case is the NRS 1, a tentative spec for the design of reactor pressure vessels and associated components. It was issued again as a tentative standard with the specific purpose of revising it every 6 months, if need be. So, some of the spec writing branches of the government are looking forward and moving along, and not using 1886 documents."

Dr. D. Westrope, of Lockheed, asked that Mr. Klein explain the difference between a reliability test and a qualification test and tell what a reliability test accomplished that a qualification test did not.

Mr. Klein responded, "Actually, I mentioned a group of tests of which the reliability test was the eventuality of all the prior testing. The QUAL test and the acceptance test have a very definite place in our series of tests. Finally, and hopefully prior to the first use of the equipment (when it is ready to be delivered), we have a reliability test which is often called the demonstration test. At this point you attempt to meet criteria that are tied to the risk of your program. In a very low-risk program, the demonstration test could be an environmental test simulating the environment of the use on a repetitive basis, to prove that your hardware can stand up for given periods of time.

"On the other hand, we are involved every day with programs for the Space Systems Division of the Air Force where there are only two, three, or perhaps four flight vehicles. You can't go into a very long period of demonstration testing, so you must revert to high risks or risks that are commensurate with the objectives of your program. Understanding these risks, the reliability engineer, in conjunction with the environmental people, designs a test that can be labeled a reliability test. It should accumulate all the environments of the equipment from its birth to death. It is truely a demonstration of capability, with the amount of repetition being a function of the risk."

Mr. Westrope asked if this wasn't done in QUAL testing.

Mr. Klein indicated that it wasn't, since one could hardly fly a piece of hardware which had experienced long repetitive testing.

Mr. A. Oppenheimer, of Walter Dorwin Teague Associates, suggested that a good deal of the criticism of specifications is due to the fact that a spec is often abused. He believed that a spec, if used properly, was a very useful tool. Initial performance requirements were only best guesses, and the design engineer should see to it that the requirements were updated as the state of the design advanced. It

had been Mr. Oppenheimer's experience that government agencies were very cooperative in attempts to obtain deviations if the proper approach was used.

Dr. Mains commented that, in many cases, the lawyers or contract officers tended to take the spec literally, but, if one could get behind them to the technical people, things could generally be straightened out.

Mr. E. Stoops, of North American Aviation, had had a different experience. His request for a change was denied by the legal people even though it was supported by reliable field data, and the technical government representatives were in agreement. He wanted to know how one overcame that.

Mr. Klein answered, "In general, specifications are guides. The applicable documents are written in the work statement when it goes out to the contractor. In every case, the contractor responds to the work statement and contractually agrees to do a job based upon the applicable documents. If he desires to make statements in his proposal as to how he intends to change the specification during the program, he can do so.

"In the specifications that we have written covering the programs at SSD, the intent is that they are merely guides to the contractor by which he responds with a working plan specification of his own. If his plan is agreed to, it becomes the legal document, not the MIL spec. If the contractor does not take exception to the spec at the time he makes the proposal, that is his responsibility.

"We recognize that some of the specifications for our vehicles represent the best knowledge of the state of the art at the time they are written. As a result of ground testing, PFRT's, and so on, we refine the specification. We refine it again after the first series of flights and again after a later series of flights. There are any number of missile programs in which the specifications were delineated several times during the flight test program. I think that the contractor should bear in mind that a lot of the responsibility rests with him to define, in his proposal, how he intends to use the spec."

Mr. J. Wolfinger, of AiResearch, said that his company had heavy Ground Support Equipment (GSE) going from LA to the Cape with specifications for wide extremes of environment such as snow, rain, sleet, and vibration. The equipment weighed as much as 1 or 2 tons and was going to be used in a MIL standard clean room in a sheltered environment, and was to be treated better than usual. He asked the panel to discuss the relative merits of qualifying this GSE hardware in the shipping packages rather than having to design and test the equipment itself to these adverse environments.

Mr. Ruby responded, "Much of the gear we design at Motorola is similar, ground base equipment in a reduced environment. We interpret the specifications, with the aid of the contracting agency, to mean testing in transit cases. If you have a shock and vibration shipping specification, then the particular piece of susceptible gear should be tested in its case, not solidly mounted to the exciter. As far as I can see, there is absolutely no point in testing a delicate piece of equipment to a road shock which, hopefully, it won't see."

Mr. R. Daniel, of RCA, concerned with satellite design, asked whether qualification testing of components should be covered by a general specification or by an individual specification for each component, based upon an estimate of what the environment will be. "In many cases, the final layout of a satellite design does not become firm until very late in the game. Therefore, should one test to the most severe condition for every component or simply arrive at some arbitrary number?"

Dr. Vigness said that it is practically impossible to write a sufficiently broad specification to include components as its objective. "As nearly as possible, general specifications are written to cover the field conditions as they might exist under an average case, in order to make the overall equipment survive in service. There is another type of specification which is written for an individual part and is generally designed by the project officer responsible for obtaining that part. He looks into the situation in more detail and usually produces a parts spec quite different from the overall equipment spec."

Mr. P. Perry, of A. C. Spark Plug, supported the arguments that testing equipment to environments it will never see is silly and that the fault lies with the contractor for not interpreting the specs early and registering his objections. In general, Mr. Perry thought military specs were pretty good.

Dr. Mains thought it might be worthwhile for the panel to consider the question of how much preliminary engineering it was justifiable for a company to do in order to search out, ahead of time, the exceptions they need to take to the specifications.

Mr. Ruby presented another problem that exists, particularly in the proposal stage. "At a bidder's meeting, a statement is often made that this will be the last opportunity to ask questions. This develops an almost pathological fear among the people building equipment, to go back and find out what they really mean. In reality, if you call somebody up, you can get the information."

Mr. Newhouse, of Marquardt, thought the problem really stemmed from the procuring activity, in that they would just arbitrarily spell out requirements from some existing spec that the equipment would not see. He felt that it should be determined at the start of the program what the equipment environments were.

Mr. Klein said that as long as the procuring agency does not know the details of what the contractor was going to do, the equipment being procured would be required to be tested to any environments that it might see. The contractor had the right to stipulate in his proposal that he would protect the equipment during certain specific environments and, on this basis, the procuring agency might waive the tests for those environments. In all cases, Mr. Klein believed in the environmental profile as the contractual obligation with the answers to be given by the contractor.

Mr. P. Moore of Redstone Arsenal recalled that, during the war, there were 6 x 6 trucks which were supposed to go 20,000 miles before they needed any major repair. Apparently, because of rough treatment by personnel, they only lasted about 1000 miles. He suggested that if our specifications really showed these environments we wouldn't have a problem of shipping, packaging, or anything else.

Mr. Moore said that at the Missile Command they did waive certain specifications when it was justified by the contractor.

Mr. J. Barrett, of Watervliet Arsenal, remarked that to some people, particularly equipment vendors, it seemed obvious that a particular spec was not applicable. He wanted to know why this was so when it was evidently not obvious to the user. He suggested that the user should specify that the equipment shall perform certain functional duties and let the vendor guarantee that it will perform.

Mr. Ruby responded, "Concerning why a vendor says a spec is not applicable, I think, facetiously, it's like asking the price of a car. It depends on whether you are buying or selling. If you're selling, you want to get the best price with the least restrictions. If you're buying, you want the most restrictions.

"In answer to your other question, there is a design spec that we at Motorola have been working on. It was described at this meeting, in a paper by Mr. Baum, and it is particularly concerned with thermal requirements. I realize that there is, at least, an order of magnitude difference between shock and vibration and temperature, but I think our spec tries to do what you're after.

"It prescribes ground rules to tell the equipment manufacturer exactly what his equipment will do thermally. This is not a Go No-Go test; it gives him a profile. He knows not just that it will pass some MIL spec, but what it will pass. He can then fit this information into a specification profile and see whether the equipment is good for it. It is a tremendous selling point for the manufacturer."

Mr. Lynn of Lockheed said, "I think probably there was a dig at what I had to say on how the manufacturer knows that a specification is wrong. I guess I come by it by 30 years of experience in packaging and handling engineering. I'm sure that most manufacturers don't have the benefit of the observations that I've made over that 30 years, and often are not acquainted with the tests which go with the specification of any one package. I'm quite sure that people who write the specifications are not altogether sure of what goes with the specification of any particular package.

"In section four of every specification, the packaging tests are specified. It's not easy to get a change. When you are testing, you should recognize the fact that the tests on the packages are sometimes more severe than will be encountered during the use of the part. Really, the environments of handling, transport, and storage are more severe than the operational environment, especially in shock. This is something that is really important and it should be taken more seriously by the government, I think, than it has been up to this point."

Dr. Mains commented, "I'm sure that there are some people here that either are presently in one of the military services, or have been. A couple of weeks ago there was an Army Ordnance meeting in San Antonio with military people there. The uniformed men were stressing that equipment must work. They're sick of field failures. For example, last winter on a base in Alaska, one morning there were only two vehicles on the whole base that would start. When they did get them started and rolling the

tires fell apart in chunks because it was 70 degrees below. You can't tell those boys that the specs are too severe. They'll say the specs are not nearly severe enough. They have to have reliable gear."

Mr. E. Stelly of Texas Instruments referred to previous statements that the actual environments were not as severe as the spec requirements, and to Mr. Ruby's comment that after the contract had been obtained they could not do the test job. He said that this did occur a lot of the time because people didn't take the time to contact the test department. His group was trying to inaugurate a program whereby all proposals would first be routed through their area so that they could check to see that all requirements could be met. If they could not be met, checks would be made to see about buying equipment or subcontracting the job locally.

Mr. P. Marnell, of Technik, Inc., asked, "Are contracts currently being written so that it is the legal responsibility of the contractor to produce an item which will perform satisfactorily in service operation?"

Mr. Ruby replied, "I think this is really a matter for the lawyers. If you contract to build a piece of equipment that will live in its real environment, this may take 5 to 7 years, the specified mean time between failure. Is the contractor going to wait to get paid until the end of this time to see if it has failed, or is he going to get paid immediately and then give the money back? You might tell the man that if it doesn't perform he won't get any more contracts. I would think that it would be an impossible situation to expect the builder to guarantee it for life and then pay back if it doesn't work, or that type of thing."

Mr. Klein commented, "While at STL and now at Aerospace, we had a great number of contractors working on each of our weapons systems and in no way were we able to tell the associated contractors that each of their equipments should function in a normal manner in the use environment. Instead of this, we wrote a requirement that the total system should work in a given manner with one associate contractor becoming responsible for the system. The remaining contractors were responsible for delivering their equipment to the systems engineer who was going to fly the entire vehicle. Each has performance requirements on each of the pieces of equipment that he delivered, including one-shot devices, and it was his responsibility to devise a test program in answer to his requirement for use. He delivers his equipment to the integration contractor and, from that point on, he is paid.

"In the final evaluation of the vehicle, the systems contractor flying the vehicle will not accept responsibility for a failure if it's not his equipment. Essentially, in the complex programs that we live with, you can't expect each man to guarantee the performance of his single piece of equipment, because he has no idea what the associated contractors will do to his equipment after it is delivered."

Mr. M. Christensen, of Aetron, recalling that others had commented about problems of guiding the lawyers, said that, in large bid proposals, those who knew what the problems were, bid accordingly. They bid high, throwing in lots of contingencies, because they could legitimately anticipate problems of shipping, handling, and so on. But a lower bidder, who didn't know or care about those things, was likely to get the award. Mr. Christensen asked what could be done to help the lawyers to see that everybody was bidding fairly on the same big requirements.

Mr. Klein responded with a question. "How many fellows who've worked on equipment have ever read the proposal for the equipment they are working on in their own company's house? I think we really ought to refer back to the proposal and see what our bosses, our systems engineers, our facility people, and our test people are committing us to. If they are committing us to something we can't do, I think we should immediately demand that we get better equipment and change the spec. You can't read the proposal 8 to 10 months or 2 years later and discover that you can't do the test. It is the responsibility of each test engineer to know why he is doing a job and to understand the environment.

"My suggestion is that when you work on complex, expensive systems, you should understand what you have by looking at the proposal. It will give you a real sense of strength when you have to deal with the procuring agency and your own lawyers, as well as ours."

Mr. Johnson, of Atomics International, asked, "Is it feasible to give the contractor some feeling or some understanding of the real environment that a system is going to see? Our experience with a satellite is that we were given a simulated environment for shock and vibration to which we designed and, as the product evolved, the company issuing the spec found that it was not adequate any more. Due to some peculiarities, it was not completely applicable to the design that we had. Would it be feasible to put in the spec a general outline of how they arrived at their simulated environment and also some of the assumptions that

they have used, so that we will have an indication that possibly the test has to be more severe or if the tests we are using are not good for our design?"

Dr. Vigness answered, "Generally, it is not feasible to give all of the information which is background for deriving a specification. If a company or an organization wishes to learn this, it really has to find it in the literature, such as in the Shock and Vibration Bulletins and the various reports on field measurements.

"The legal aspect is a very difficult problem; I don't know how we can keep it from sometimes controlling the amount of effort we have to spend on these things. The first step necessary in getting a change, however, is to get the three parties—the test people, the procuring agency, and the contractors—together so that they know what should and can be done and what the apparatus can really withstand. After that has been thrashed out, then they will have to go to the legal department and see what adjustments can be made. There really isn't any way around that, as far as I can see. The legal things have to be taken into consideration, but we surely shouldn't let them control the fate of a project.

"Many times we will get new information as time goes on and, in almost all cases, the specifications must be changed to correspond to the new knowledge that we have. That can be done, but it may require adjustments of contracts which is messy to do. It would be much better if we could, as was stated earlier, just make a requirement that the equipment survive the expected environment. That gets to be kind of impractical in the real case because we have to send it through the expected environment to know whether it is going to survive. It would, however, allow us easily to make changes in any specification."

Mr. Nankey, of GE, was concerned with the manner of maintaining applied vibration to equipment with multiple mounting points. "There has been a lot of controversy as to whether minimum input should be maintained at each mounting point or whether some sort of an average should be used for a maximum. My own opinion on this is that an appropriate average like the average absolute value of specified inputs should be used, and that this would accomplish the intentions of the specification. I've heard of a lot of effort toward maintaining the least responding input point at the specified level. When this falls on a nodal line or a nodal surface, it can cause all sorts of trouble."

Mr. Lindner felt that the problem of different measurements at different input points was a result of fixture problems and that improved fixture design would help considerably.

Mr. Nankey said, "I'm thinking of the cases where we're dealing with equipment that's large enough and working with a specification with a great enough frequency range that we cannot avoid fixture resonances."

Mr. Klein responded, "We know that this is not an unrealistic problem. Our environmental studies group not only handles the specification writing, but we have one of the men from the group go out to the contractor's plant and monitor the contractor test. This man has the same kind of experience that you do; he is a test engineer. He would have perfect raport with you and the solution would become a logical solution between you and him.

"There are many measurement points at the input of a vehicle. You can't take the minimum; you can't take the maximum; you work out something that appears to be the average for the particular piece of hardware that you are working with. In all cases, it becomes a logical solution and you play a very important part in the area of testing. You need the same kind of person to work with as the one who monitors your test. We would have no argument with you."

Dr. Mains commented, "A few weeks ago, I heard a man give a paper on an averaging device for monitoring the input on a shake test. He weighted each monitoring signal by passing it through a potentiometer and introducing a different phase lag on each item. Then he proceeded to add up the instantaneous value and divide by x or whatever the number was. This was his monitoring level. He was kind of surprised that he wasn't getting very good control of his test. Some people thought it was nice to do that. I was amazed."

Mr. L. Pulgrano, Grumman Aircraft said, "I'd like to hear some comments on the application of motion input testing to equipments or space craft, with fairly large impedances. It's common to use the envelope of measured vibration data on a missile flight as an input to a relatively large space craft. Inherent in this is the tacit assumption that the launch vehicle impedance is large relative to that of the space craft. In the test, one might find amplifications on the space craft of 5 or 10 or even higher above the input motion that is representative of the maximum measured levels during flight.

Next, in many cases, an envelope of measured response levels on the space craft is taken during the vibration test and used as an input to the equipments. Again, the assumption is that the space craft impedance is infinite as compared to that of the equipment. One could logically follow this and let the equipment man take inputs on his equipment and give them to the component man. You could build up extremely large amplifications in this manner, much higher than anything that is ever measured on any kind of a launch vehicle. I'd like to hear some general comments about this kind of test and also, perhaps, some specific comments on what kind of test could be used in its place when we're dealing with a structure that is known to have significant impedance."

Mr. Klein remarked, "You neglected to say that sometimes a space craft has an attenuation of 10, 20, 30, or 100 times. If you put in 10 or 20 g's, the package may only see 1 or 0.5 g."

Mr. Pulgrano agreed that this was also a problem.

Mr. Hahn said that they had run into this too and, in one of their vehicles, they had to include the launch phase as well as the airborne phase to develop the envelope Mr. Pulgrano spoke of. Mr. Hahn continued:

"We also investigated analytically, by chopping off part of a missile, trying to devise some kind of mechanism to get an effective mass to cover, at least, the fundamentals in a missile lateral bending mode. Of course, we cannot correlate all the frequencies, but we can take a piece of this and, at least, approximate the impedance of the whole missile."

Mr. Pulgrano inquired, "Then, essentially, you apply a motion input to the structure on which you've mounted your space craft? Do I understand you correctly? Do you just add a little bit more of the launch vehicle structure and then apply a motion input?"

Mr. Klein answered, "If you make the measurement at the interface to the space craft during the launch, orbit, or whatever portion of the flight you are interested in, and you use that interface portion of the forcing vehicle, it's as close to an engineering compromise of a real life problem as you can get."

Mr. Hahn interjected, "This is just what I was getting at with this added mass or impedance. In other words, if you wanted just to test a nose section and you could get some kind of equivalent impedance, then the interface between the nose section and whatever is following, would be the point at which you would apply the motion."

Mr. Klein added, "It might be valuable to make apparent mass measurements with dummy packages in the payload. I think these are R&D problems that you'd solve during the early stage on the work with your structure and the location of the packages. Many times the apparent mass measurement will show you that you really don't have a severe problem, since it is very likely that your structure acts as an attenuator. In general, payload structures are light, quite flexible, and provide a great deal of attenuation."

Mr. R. Colyer, of Boeing, said that, in the MINUTEMAN program, hardware was being delivered at the same time that they were running qualification and other type tests. Under pressure of a cost reduction program throughout the Air Force, they were considering using field experience on some items as a substitute for QUAL tests, assuming that there was a one-to-one relationship between the field environment and the QUAL environment. He asked for comments from the panel.

Mr. Klein said, "I don't feel there is a one-to-one relationship between the flight environment and the QUAL environment. Since we live in a world of transients, how do we know where we stand at any particular time? What's the safety margin on a one-to-one relationship? We would like to see a qualification test on a single piece of hardware at some definite safety margin and to see each piece of flight equipment acceptance-tested at somewhere near the flight environment. We must have assurance that it will work at the flight environmental levels."

Mr. Colyer mentioned that they had in mind such things as transportation vibration, sand and dust, sunshine, rain, and that type of thing. Representative equipment had already been delivered to the field under those conditions and had arrived in good shape.

Mr. Klein said that he had misunderstood, and that he was thinking of components in particular. He agreed that the substitution may be alright if it can specifically be shown that prior equipment has survived. He mentioned that, in some cases, expensive tests are omitted on new hardware where very similar tests have already been performed.

Mr. S. Baber, of Boeing, commented, "You're in really dangerous water when you

run QUAL tests and flight proofing tests after the fact. Usually, if you have any failures in the laboratory, you start looking at the laboratory test. We have had quite a number of examples where, by adding some of the missile structure as a part of the specimen, we could eliminate a large number of failures. We've had items that, alone, would last only for about 15 seconds in a random vibration environment, yet by adding the missile skin and the missile structure, the item would go through a complete 1-hour test.

Mr. Klein said, "That's the purpose of QUAL testing, too. I think you answered it very nicely."

Mr. I. Sandler, Autonetics, said, "We were involved in a test for which the specs were written before there was any available data. After the first few flights, we felt that the spec was rather high and we attempted to get it changed, at least to reduce the time of vibration. We were met with a mathematical dissertation showing that it didn't make any difference and, therefore, there wouldn't be any need for a change. I wonder if we were unreasonable or if this was an unreasonable approach by the people who wrote the spec?"

Dr. Vigness responded, "Generally, when you try to show how good a piece of equipment or a component is, you try to do it in terms of the probability that the item will pass a certain level of vibration or shock. One item might pass; another item might not pass. Your trouble may be that you don't have very many items to work with, so you can't really get a statistical study of the thing. If one works with only one or two items, and if one can guess some sort of reliability or fragility curve for the particular items with a reasonable exactitude, then one comes up with some figure for the tests. This figure will be somewhat higher than the actual environment, because you don't want only a 50-50 chance of a thing operating. You may want a 99 percent chance of it operating. This means that you generally have to test to a higher value than you might expect in the field."

Mr. Sandler pointed out, "We weren't trying to lower the value; we were trying to reduce the sweep time. The same agency that wrote the specification also had published a paper showing that the sweep time made no difference. The ultimate result in the test was that it did make a difference if we vibrated it for a shorter period of time. The hardware had a very hard time passing the original spec, but it did finally pass the revised, what they called a minimum confidence, test and it did prove to be a good piece of hardware in flight."

Dr. Vigness concluded, "I would expect that, if a theoretical study showed that it didn't make any difference and an experimental study showed that it did make some difference, I would prefer to go by the experiment and try to get a sweep time which would be long enough for the most severe condition."

Mr. R. Hunt, MSFC, commented, "It is implied that there is a particular environment for a particular piece of equipment. I think most people have equipment that is being used on several different vehicles. They are probably having trouble qualifying this piece of equipment in one environment, yet, maybe it's living happily in all of the others. If you're not qualifying your equipment, it's not being sold. We are all on the same side. Whether a vendor's equipment works is as much my problem as his, except that the way it affects us is slightly different. I'm aware of this difference.

"The best answer is this closed loop, the feedback. Now, from our operations standpoint the group I'm associated with does write the specs at Marshall. Anytime there is a test being run, if the vendor will inform us, we will have someone at his plant. This accomplishes two things. First, it helps close the feedback loop, since we know what he's doing. Second, he gets to talk with us and find out what we would like to see done. In general, this close liaison provides the mechanism for changing a specification if it is necessary."

Dr. Mains, in order to bring the discussion back more closely to the session topic, asked each of the panelists a question. "Suppose that we have an environment that we can measure and define as precisely as we care to do, what, then, should be the relationship between the test that is to be performed and the environment we have measured? Should it be more severe, of shorter duration, or what? Mr. Ruby likes the thermal end of things, so in a thermal test what should that relation be?"

Mr. Ruby answered, "I think the thermal aspect is somewhat different from the shock and vibration environment in that it is easier to specify. It may not be more accurate, but, at least, it is easier. For example, we say that an electronic component will not work well above a particular temperature. This, then, is the endurance limit or qualification level, whichever you wish to call it. If we can go to our reliability people and have them run a few tests, we can get accurate numbers. We can

then apply these numbers to the particular piece of equipment and, indeed, test it to the real environment, or test it until the most critical part gets as hot as is allowable. This is the worst environment that this equipment can stand. We can then plot our entire profile.

"For the thermal aspect, the real environment would be perfectly satisfactory. I think you could say with some assurance that, if we knew exactly what temperatures, pressures, and so on, the equipment was going to meet, we could test it accurately in the lab with a fair confidence that it would perform throughout its life."

Dr. Mains then asked Mr. Lindner, "If you knew the ground transportation and handling environment as well as you would like to, would you then make the tests more severe, less severe, or what, with relation to that measured environment?"

Mr. Lindner responded, "If you knew precisely what the environment happened to be, how would you simulate it in the laboratory? Generally, I think that, if you knew what the environment was, it would be very good for design purposes, but the way you would actually test your piece of gear would likely produce something entirely different from the stated environment. For example, the field vibration environment is probably omnidirectional, whereas your simulation equipment is usually unidirectional; however, you should have some correlation between the two.

"To answer the question, let's consider the railcar humping problem. If I wanted to simulate this in a laboratory, the information I would need includes velocity of impact, how the car was loaded, and the type of cars against which to impact. In my test, I would go higher than the mean value of the velocities in rail transportation because I want better than an even chance of having my equipment survive. I think this would hold true for any environment. I would overtest rather than take the average value."

Dr. Mains' question to Mr. Hahn was, "From an analytical point of view, is there any advantage or disadvantage to be gained in making the test more severe or less severe than the measured environment?"

Mr. Hahn said, "I think this time I will defer back to Lockheed and say that Mr. Blake's decision theory is one of the approaches that seems to hold a lot of promise for giving us these answers in the future. In such a theory, things like the original design, the quality control, the manufacturing, and so on could be considered and from all this input we would derive our test criteria. The field of design analysis is somewhat bare. In the flight regime, we are pretty well along the way in defining the types of gust and wind shear which occur in the atmosphere along with their associated probabilities. I would like to say to the Army, that we should like very much to see something comparable to this in the description of terrain, so that we can design ground mobile systems with the same confidence that we think we do in the airborne case. If we were going to operate in the Sahara, or wherever Dr. Brierly had figured he could give us some idea of what the terrain would be, we would take something that was already available such as Aberdeen and find out what combination of tracks there would give us the required simulation for our operating conditions. This, then, would be a basic specification for design of ground mobile systems and if, then, the airborne environment were superimposed on this you would have what you could consider your fundamental specification, your source environments. From this you would deduce the equipment environments, the shears and bending moments, and so on, that the man on the drawing board would like to see. Today he tries to design to load factors. How do we vibration-test to load factors? In fact, what do they mean? How does decision theory enter into this? These are questions I'd like to see answered. Once we get methods of this type, then we can design our vehicles. Also, with a proper application of probability theory we could devise a test that the normal vibration machine produces which would have associated with it a confidence level and all the other regalia that quality control requires. Within that level of confidence you could perform your design and hope that your test would similate what you had designed into your vehicle and equipment."

Dr. Mains said, "Before I pass this microphone to Phil Klein I'd like to add another condition. We know what the field environment is and we can measure it, but we also know that at least a goodly proportion of the failures that occur as a result of a motional environment are fatigue failures. We know that under the best conditions of laboratory testing you get at least a decade of spread in fatigue data; that you get two decades of spread if you let any corrosive atmosphere or liquid come in contact with the specimens; and further, if you let a little abrasion or surface maltreatment take place, you can then get three decades of spread. So, we have an environment that's defined within 98-percent confidence of 95-percent accuracy, but

of risk that you wish to take can be indicated in the safety margins; it can be indicated in the duration of the reliability test."

Dr. Mains asked Dr. Vigness, "Now, Irwin, from the standpoint of the fellow who has to sit inside the turret of a 5".38 and aim it, or from the standpoint of the tailgunner in a bomber who has to use an electronic gunsight, or of a fellow who stays on the ground and tries to point an HONEST JOHN missile in the right direction, what do you think these things should be tested for? Just the field level, or more, or less, or what?"

Dr. Vigness responded, "I think Mr. Klein here has stated the thing so nicely that I would not state it much differently, as far as that phase of the question is concerned. I was irritated by the assumption in the original question that we know the real environment. The environment that any piece of equipment sees is never something that you can define until after the item has lived its life and you have followed the environment all through that time. There is no such thing as a predictable real environment in detail. You cannot make any test which corresponds to a real environment because they are all different, therefore the tests you devise must be a conglomorate of the real environments. Mr. Klein mentioned this when he was referring to an envelope of conditions. There are objections to envelopes of conditions too, because they don't take into account impedance and so on. That was brought up by some of the talks yesterday. Inasmuch as we can't predict the real environment, but can only have some average or fiducial limit type of environment expressed on the basis of probability, perhaps, then we will devise a certain level for a test based on all probable environments that might be encountered. The tests which might be performed on the vehicle then might be quite different from any specific environment. Many times it has been suggested that we take tape recordings of vibrations of missiles and play them back through the vibration machines. That would not be a good test generally, because it would only be specific for that particular condition. We have to have something which is representative of an average condition."

Mr. R. Roberts, of GD/Electric Boat, asked, "What should be the relation between the measured environment and test in the case of the shock environment due to underwater explosion? I'm thinking particularly now of hull-mounted equipment such as valves, machinery, such as turbines, electronic equipment, and the like."

Dr. Vigness answered, "There have been an enormous number of measurements made of the shock conditions that these types of equipment encounter onboard ships. In submarines, particularly for items close to the hull, our shock machines cannot deliver a shock which is as great as the shock that might be experienced by the submarine, and which would seriously damage the hull of the submarine. We cannot produce shocks as great as we would like without destroying the machines that we use for shock testing, because they are made of structural steel. The ship is made of structural steel, but presumably it doesn't have to withstand more than one or two of these shocks, whereas our shock machine has to withstand many. So we go to about the maximum amount of force that the machine can deliver repeatedly. For equipment which might be mounted inside a surface ship, our shocks could probably be somewhat less severe than would be normally specified, but we usually can't say where a particular item of equipment is going to be located. Inasmuch as we don't like to make a piece of equipment only suitable for a particular location, we have it tested to the general specifications for any location onboard ship."

Mr. K. Johnson, of K. W. Johnson & Co., said, "We would be very happy, since we manufacture shock and vibration control systems, to work with the MIL specs. Generally our trouble is that the MIL specs are re-interpreted by each project engineer of each company, and safety factors of one form or another are always added. If this were done on the basis of an engineering evaluation, and the service need were greater, then the MIL spec requirement should be increased accordingly. So often it's just a number as far as the engineers are concerned. If it's tested at 5, 10, or 20 g, it's simply doubled to provide a safety factor. As a result, we get as many specifications as there are different companies and, sometimes, different engineers. You can see that we have a problem in trying to come out with any type of a standard procedure so far as designing a shock system to satisfy more than one particular application. For example, at the present time, we have, for two different companies, entirely different mounting systems for the same type of system going into the same location and meeting the same spec. It is a result of different interpretation by the individual project engineers. How do you answer these problems?"

Mr. Klein answered with questions. "What kind of vehicle went on top of the vehicle you

are talking about? If you're talking about a first stage with a certain kind of suspension system, what went on top of that first stage, what went on top of the second stage?"

Mr. Johnson said, "This is a normal electronic type of gear that would be protected from shock under an aircraft condition, in this particular case. It's not a missile condition."

Mr. Klein responded, "Even so, perhaps the vehicle is being used for a different kind of function. For example, if you design a certain system for use in a ballistic weapon, and then you decide to use this ballistic weapon as a space booster, the device that goes on top of the space booster, which was real clean (perhaps a nicely shaped satellite for orbital conditions), suddenly grows to some monster that looks like the MERCURY package with a rocket lifesaving device on top of it. If you are looking at the lower uses of the equipment installed in the first stage, the environments it sees may change by a magnitude of 10 or 20 times. This may happen even in truck equipment where you use one truck for carrying boxes of food and another to carry a missile launching device. The suspension system installed for certain pieces of equipment really needs to be changed, even though the configuration of the vehicle is the same."

Mr. Johnson agreed in principle but said, "As I qualified it, we're talking about rather simply defined systems. The same type equipment goes into the same space, performs the same function, only it is provided by competitive, different companies. We get such a wide variation, in the interpretation of the MIL specs, as to what environment the equipment is to meet. The shock requirements or vibration requirements often go through several different groups in a particular manufacturing company, and each group feels that they need better safety factors. They say, 'If this number is x, we'll make it y, if it's y, we'll make it 2y, since we want to be more safe in our design.' In so doing, it changes the whole design of the mounting system, making it much more costly and everything else. It's the arbitrary type of change I'm thinking of, not one resulting from a real engineering evaluation."

Dr. Vigness said, "It seems to me that in what you are talking about you have two different groups designing something to serve the same purpose. These two different groups approach the thing somewhat differently; they buy some material from you to go into their products. Inasmuch as they have designed the thing using a different approach, but for the same end purpose, it's not very likely that they will have the same design all the way through. They'll even have some difference in properties, depending upon what they thought was safe and what was not safe. I don't think there is much you can do about it, except just try to please both of them."

Mr. J. Brunn, of Sylvania, said, "We have a requirement to shock test an equipment rack weighing over 1000 pounds and standing about 7 feet high. The requirement is to shock test this rack in its shipping container on a shock spectrum machine. To my mind it would be more logical to drop test it on a concrete floor; it would actually simulate the environment more realistically. I was wondering if maybe this isn't a misuse of the intended purpose of the shock spectrum type shock machine."

Dr. Mains asked whether he was sure about the end use. He thought it sounded like a stowed container test.

Mr. Brunn said that it was merely a shipping test in order to simulate the shipping and handling conditions between the factory and the permanent installation, a protected installation.

Mr. Lynn, of Lockheed, commented, "The problem of vibration and shock to electronic console equipment is an industry-wide and growing problem which is far from being solved. One of the reasons that we're having difficulty is that no electronic equipment manufacturer will give us fragility numbers for the equipment. I imagine what Mr. Brunn has run into is an effort to find some of the numbers that are associated with the actual equipment, because we are having to take these things into laboratories at the time of their arrival and realign. It happens maybe two or three different times. Quite often, the equipment is in use at one point for 3 or 4 months, and then it moves to another place and is used 3 or 4 months. Each time it goes through a laboratory for alignment; sometimes it is scavenged; sometimes it is damaged in the laboratory. Sometimes we find that there are stamps applied that it has been inspected and that it was OK or that pieces are missing from the equipment. I think this test is an effort to discover whether or not the equipment is misaligned by shipping. I wish there was more of it, because I think, in the long run, the taxpayers will pay a smaller price for arriving at the point of 100-percent reliability."

Mr. Brunn said, "I have no objection to shock testing the equipment, but my understanding would be that it would be more logical

to shock test it on this machine in the unpackaged condition. I think this would be more of a controlled test and would give you a better idea of the structural integrity of the design."

Dr. Vigness agreed, saying, "It seems to me that a drop test on, let's say, a plastic pallet or a drop table type machine, would not normally be specified for testing packaged equipment, and that the normal tests, which are to drop it from certain orientations and in certain ways, could just as well be done. However, that is a matter that you have to work out with the persons who you make your contracts with; if they have reasons for doing it one way and you object to them, then you just have to find out what those reasons are and show them why something else might be better."

Mr. Baber, of Boeing, asked, "Is it either desirable or possible to list and define the flight or the service profile, as best we can, in the specifications and, rather than listing the tests, to list the objectives of the test, such as to simulate the failures and not the environment?"

Dr. Vigness responded, "If we have an opportunity to have feedback as to the types of failures, we can adjust the tests gradually over a period of time so that they will produce the kind of failures which actually occur in service. This is a luxury which we, generally, can't have when we are developing limited numbers of new equipment. In something such as packaging commercial equipment, there is feedback as to what types of things fail; so, they adjust their packaging techniques to correct those things and eventually get an economical package. Tests are made which will correspond to the field condition, but they may not duplicate it. That was the way the Navy light-weight shock machine was arrived at by the British. They made tests on a machine which, more or less, duplicated the types of failures that they were having on their ships. They did not take a large number of measurements and then make a machine which would correspond to some average of those measurements. But it requires feedback."

Mr. Klein added, "In general, we collect a great deal of flight environment data on missiles such as ATLAS, TITAN, THOR, MINUTEMAN, PERSHING, and on conditions that you might meet on second stage vehicles such as AGENA, ABLE STAR, and so on. We attempt to get data on various satellite configurations. Now, having all of this data, when a new program comes into being, we look at the configuration, we look at the history of data that has accumulated on similar vehicles, if not the same exact vehicle, and, from our past experience, we design the environmental profile as best we can for this new configuration. At this point, you have no failure data even to estimate the kind of tests that you need. The work statements that come out attached to the description of this environment, lists the minimum test levels acceptable on these estimated environments. These tests are not intended to limit the contractor in designing his specification for testing. If he feels that he must test at much higher levels than the minimum levels indicated in the work statement, we are delighted to see this kind of result.

"But, generally, the answer comes back repeating the minimum levels because he is selling a product. In many instances, the levels become higher because of studies the contractor does perform and he indeed tests at higher levels to gain confidence in what he is doing. One of the forcing factors to improve on this kind of operation, which I haven't heard discussed in the entire Symposium, is the use of incentive contracts which are now beginning to be attached to programs. Now, even if we had described a minimum level that would not allow you to succeed with your equipment, even though you did prove through certain reliability, acceptance, or qualification tests that this vehicle was good, still you are probably going to have one hell of a time with the lawyers to get your incentive fee, because the final measure of the product is how well did it perform. If your money is tied to the performance, you have to have a high degree of confidence in the levels that you are testing to. So this becomes a double-edged sword that both of us have to use with real care."

Mr. Edgington, of White Sands Missile Range, asked about the development of transportation vibration tests from field data. He wanted to know how one arrives at a length of time for a laboratory test that produces equivalent damage to a given number of miles of cross-country travel.

Mr. Lindner answered, "The crux of all vibration problems is probably how long you test an item for any given condition. I'm not sure that there is a straightforward answer to this, but I have had a little feedback. A company sent a few packages on about a 1000-mile trip, and then looked at the damage. They tried to repeat the damage process on a LAB vibration exciter and, in their particular case, they came up with about an hour's run at roughly 270 rpm at the 1-g point. This duplicated quite a bit of the damage. Now in answer to your question, I can only say that you should attempt

to reproduce the damage in the laboratory, and feel your way. I don't know if there is any clear-cut method of saying 1000 miles is equal to 2 hours, or what have you."

Mr. Colyer, of Boeing, addressed Mr. Klein concerning reliability testing. He said, "For a piece of portable test equipment that has an MTBF of, say 500 hours, could you define a reliability test which would be outside the qualification envelope, or prove something that you don't get from QUAL tests which already has, say, an operating life test in it?"

Mr. Klein said, "The answer would be yes. It would be necessary to establish a group of ground rules on the amount of risk you wish to take, the probability requirements for the function of this piece of equipment, and how much money you want to spend. I think that all of these things tied together would allow you to describe a family of reliability tests for this device, depending on the amount of risk which is tied to the money. These two things together would assist you in designing a reliability test for this package. For example, if you wish to have a high reliability with a low confidence, you aren't going to do much testing; but if you want it with 99-percent confidence, you are going to do a hell of a lot of testing, and you might never get the answer. So if you can establish the ground rules that are necessary for describing the end use of this item, you can indeed draw the requirement for a reliability test. This would come from people with abilities in statistics and who stand in depth behind their reliability people. I don't think it would be difficult at all."

Dr. Mains, in winding up the session, cautioned against looking at any one phase of the design process as a black or white proposition which was to be interpreted as absolute. "Dr. Vigness has emphasized the fact that it's not the environment, but all environments, each of which is different. No particular system or structure is like another system or structure in how it responds, and no test is like another test. So instead of trying always to get down to where that eighth figure that comes off the IBM machine is precise, let's admit maybe it's just the first two, or maybe only the first one that has any meaning and see how that affects the theme of this particular meeting—the relationship of environment to specification. Thank you for coming and participating."

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Section 7
INFORMATION EXCHANGE

A COMPARISON BETWEEN A "SLIPPERY TABLE" AND A "SLICK TABLE" FOR HORIZONTAL VIBRATION TESTING

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INTRODUCTION

When the concept of using an oil film to float a horizontal driver plate on a suitable table was first introduced to vibration test personnel, it was received with much enthusiasm. This proved the answer to many problem areas associated with existing horizontal flexure or swing tables because of the inherent ability of the oil film to reduce vertical "cross talk."

Thereafter, many horizontal flexure vibration tables were discarded and installations of "slippery tables" were made using the procedures outlined by W. O. Hansen.[1] This article presents an approach to utilizing grease instead of the usual oil film as a lubrication medium and the corresponding benefits of "slick table" operation.

INITIAL DEVELOPMENT

Our particular laboratory was by no means immune to the pitfalls of "slippery table" fabrication. The usual problems of machining a driver plate to an acceptable flatness (0.001 inch T.I.R.) and surface finish (25 micro-inches) were first encountered in the shop. Although the driver table was purchased in the form of a granite inspection plate with a maximum deviation of 0.0002 inch, it was soon discovered that the leveling and mating of the table and driver was very time consuming.

[1]Hansen, W. O., "A Novel High and Low Temperature Horizontal Vibration Test Fixture," Shock, Vibration and Associated Environments Bulletin No. 25, (Dec., 1957).

As the "slippery table" became operational and was being used almost daily for horizontal sinusoidal vibration testing, the usual evaluation was performed to determine which of the many commercially available lubricants would best suit the requirements of table load, wear on the sliding member, ambient temperature conditions, and so on. As testing progressed it became evident that an oil film was being provided for adjacent test areas as well as the "slippery table." The cleanliness required by surrounding areas was a constant source of harassment as the equipment was located in a clean-semi-LOX test facility. This, coupled with the constant attention required for table leveling, erratic deflections of the slider plate, and accelerated wear, was the deciding factor in establishing the need for a more controllable type of lubrication. The natural direction was to experiment with a more viscous lubricant, grease. Several commercial, medium-heavy density, grease type lubricants were tried before determining that Lubriplate #110 (trade name) best suited the application.

An increase in grease thickness of approximately 0.012 to 0.015 inch, immediately relieved the tolerance requirements and leveling problems, and wear between adjacent members was almost eliminated.

Since it is not possible to measure the Saybolt viscosity or the kinematic viscosity of a grease, as was done with fluid oil-type products, the average power requirement to shear the grease could not be calculated as had been done

by Adams and Sorrenson.[2] The difference was not thought to be of a sufficient amount to cause concern, as in most cases it could be compensated for by careful design and fabrication of the vibration fixture.

At first the grease was applied by hand to the driver table and was distributed by driving the slider plate at low frequency. This system was later improved by drilling supply ports in the driver plate and feeding the lubricant hydraulically. The complete system is commercially available at a very nominal cost.

Figure 1 illustrates the lubrication system in schematic form. Figures 2 and 3 show details of the hydraulic metering valves, supply lines, and the drilled supply ports in the slider plate. The lubricant is supplied to the driver table while driving the slider plate at low frequency as mentioned.

extent that the lubrication is not of the boundary type. The lubricant molecule should be asymmetrical because of the affinity of a polar molecule for clean metal surfaces. Because of the thixotropic qualities of grease the lubrication is, in fact, closely related to an oil film. "Slick table" operation over a period of 3 years has established the following information:

- Vertical hop (cross talk) of the slider plate has been reduced in the frequency range of 5 - 2000 cps.

- Alignment of the slider plate and driver plate requires a minimum amount of attention.

- Wear rates of adjacent sliding surfaces have been reduced.

- Slider plates can now be fabricated from "mill run" aluminum or magnesium materials.

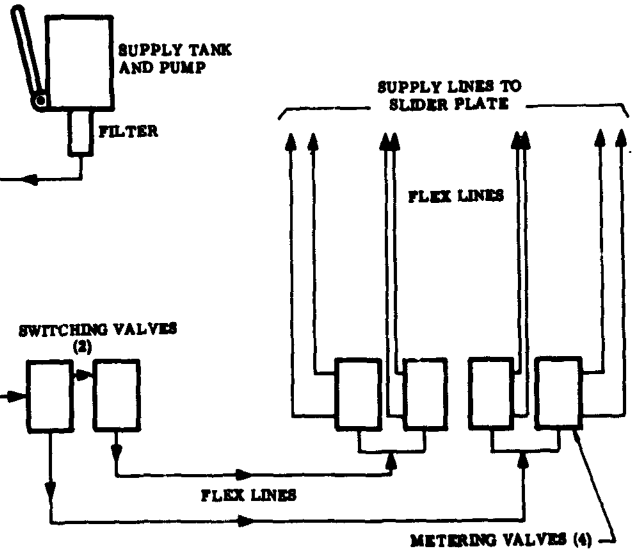

Fig. 1 - Schematic of lubrication system

SUMMARY

The lubricant used between the slider plate and driver table should be of sufficient viscosity to keep the surfaces separated to such an

- The "slick table" is capable of supporting a heavier payload than the "slippery table" without breakdown of the film.

- Exciter power requirements are comparable for shearing an oil film or a grease lubricant.

- A cleaner testing area environment has been experienced.

[2]Adams, E. C., and Sorensen, A., Jr., "An Experimental and Theoretical Study of an Oil Film Slider," Shock, Vibration and Associated Environments, Bulletin No. 27, Part IV (June 1959).

Fig. 2 - Slider plate an' lubrication system

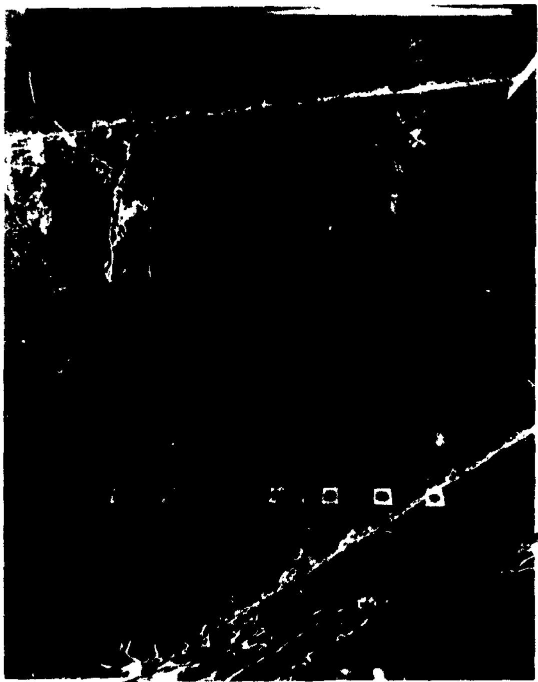

Fig. 3 - Bottom view of slider plate showing grease supply parts